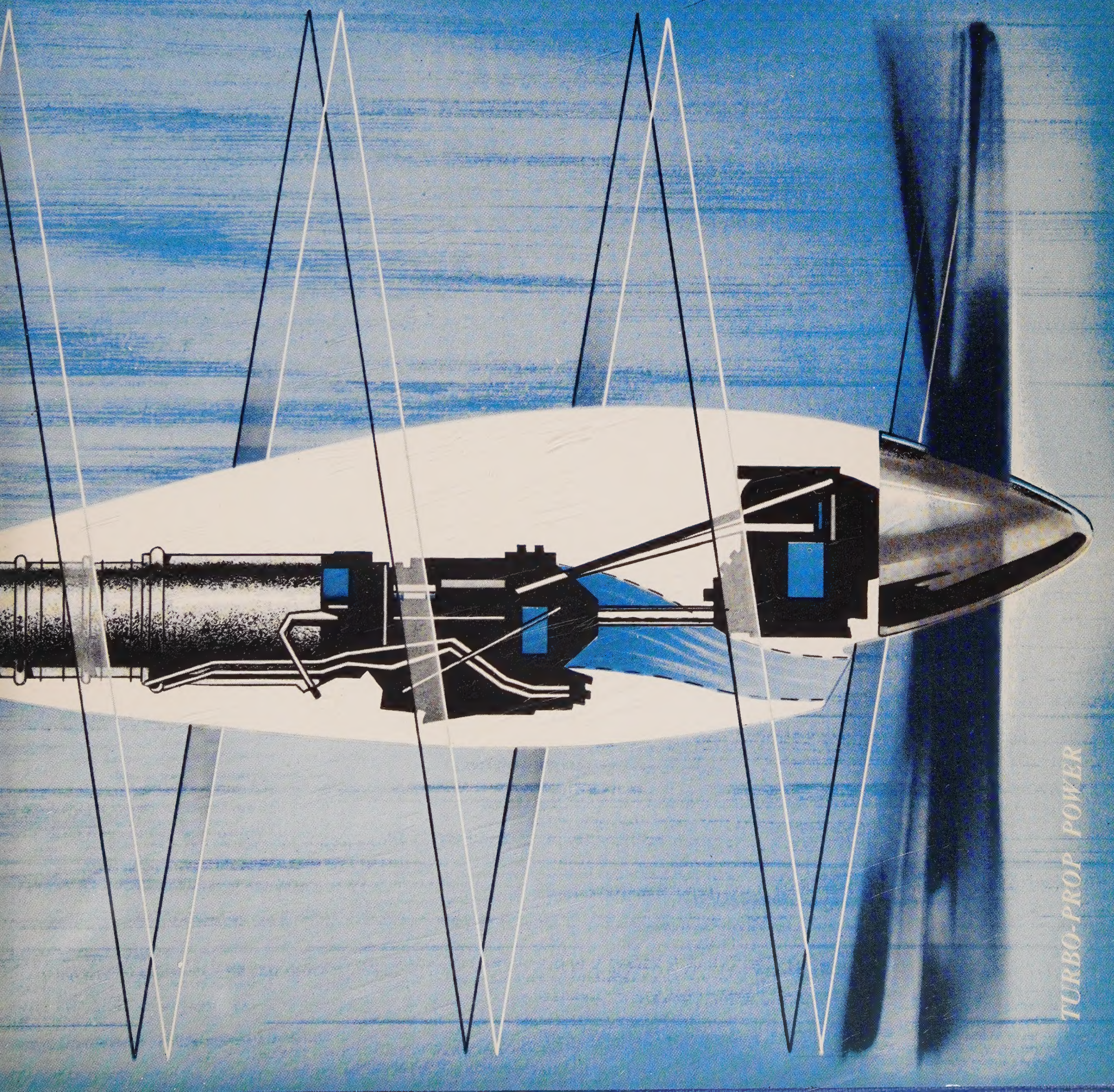
GENERAL MOTORS

Engineering

Volume 2 September-October 1955 Number 5

JOURNAL

for educators in the fields of engineering and allied sciences



TURBO-PROP POWER

Some Thoughts on Power

FOR CENTURIES man was content to live simply and unobtrusively with only three main sources of motive power to supplement his meager muscles—wind, water, and animals. But the first successful steam engine, developed by James Watt in 1769, heralded a series of remarkable developments in motive power that not only reduced man's muscular effort but transformed his concept of living. While steam power revolutionized shipbuilding, provided mechanical muscles to turn the machines of industry, and opened the way for rail transportation, it remained for the internal combustion engine to set the pace for the development of motive power in this country. And so, it might be claimed that the internal combustion engine has grown to be a critical ingredient in compounding a formula for progress, not only in transportation but other fields as well.

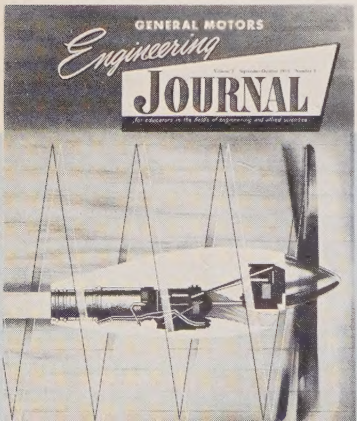
The efforts of the engineer in developing this source of motive power for automobiles and airplanes are readily perceived in the millions of vehicles which stream over our highways, and in the aircraft which wing across the sky above us. On the other hand, the engineer's efforts in revolutionizing motive power for off-the-road equipment such as locomotives, earthmoving machines, marine craft, and literally hundreds of other applications are not as apparent. Yet, these engineering feats are just as important to our total progress as are those that made us a nation on wheels and satisfied man's urge to fly.

The development of the high-speed Diesel engine is an interesting case in point. I think it is generally agreed that General Motors engineers are largely responsible for developing the first high-speed, lightweight, two-cycle Diesel engine in this country acceptable to a long awaiting market. As early as 1928, GM's research engineers began to study the Diesel engine as a possible new "work horse" in the field of motive power. By 1934 they were successful in developing the first Diesel engine which could meet the rigid specifications demanded in mainline locomotives. This engine powered the famous Burlington Zephyr on its historic run from Denver to Chicago, a distance of 1,015 miles, at an average speed of 77.6 mph, and at a total fuel cost of only $14.88. In the two short decades that followed this great event—made possible by a handful of steadfast engineers, neither bound by precedent nor discouraged by failure, but confident of their abilities in the application of engineering fundamentals—there developed a business which today composes four Divisions in General Motors employing more than fifteen thousand people. While early GM Diesels were designed primarily to provide rail transportation with a more efficient, flexible, and durable form of motive power than steam, further developments in the Diesel have filled needs for packaged power from trucks to tugboats, from rock-crushing plants to cotton gins, and from drilling rigs to submarines.

This year, General Motors will build its 100 millionth Diesel horsepower. To commemorate this event General Motors is staging a giant outdoor public exposition called the Powerama in Chicago. This Powerama is intended to dramatize the exciting story of how relative newcomers on the motive power scene—Diesel and gas turbine engines—are giving the nation new muscles to build its roads, drill its wells, plow its fields, propel its ships and aircraft, drive its trains, and turn the wheels of its machines.

The outstanding developments made in motive power since the turn of the century reflect the skill, ingenuity, and imagination of the engineer as a practitioner of the physical sciences. Yes, the engineer has helped to bring mankind a long way in a short time. And in the years ahead as new scientific discoveries are made and new materials are developed he will, in my opinion, make contributions to the field of motive power which will be even more astounding.

C. R. Osborn,
Vice President and Group Executive



THE COVER

This issue's cover design—another by artist Ernest W. Scanes depicting developments in transportation—symbolizes the engineering behind the turbo-prop engine. The turbo-prop engine is a fast maturing member of the gas turbine aircraft engine family. Because it combines the high power features of a turbo-jet engine with the propulsive efficiencies of a propeller, the turbo-prop has excellent application to aircraft in which speed is secondary to range and economy of operation. For this reason, military services and commercial operators look to the turbo-prop engine as the power source for entirely new fleets of transports capable of long ranges at speeds up to 450 miles an hour.

CONTENTS

TECHNICAL COMMENTARIES

Page

A Discussion of Economic Factors Affecting the Steel Selection and Heat Treatment for Automotive Gears, *by Vernon E. Hense and Donald P. Buswell, Buick Motor Division* 2

How to Organize and Write Effective Technical Reports, *by Ralph A. Richardson and C. A. Brown, Research Laboratories Division and General Motors Institute, respectively* 22

The Theoretical and Practical Aspects of Automotive Brake Design and Testing, *by Frank V. Klco, Oldsmobile Division* 35

ENGINEERING PROGRESS AND FACILITIES

Toroidal-Type Current Meter Improves Weld Quality, *by Paton M. Zimmerman, Fisher Body Division* 11

Applying Material Handling Principles to Containers in the Manufacture of Appliance Parts, *by Edward L. Beers, Delco Appliance Division* 30

GM LABORATORIES AT WORK

Structural Design Problems in Gas Turbine Engines, *by Philip N. Bright, Allison Division* 15

Why Bearings Seize, *by Arvid E. Roach and Carl L. Goodzeit, Research Laboratories Division* 25

ENGINEERING NEWS

Notes About Inventions and Inventors, *by Edwin S. Dybvig, Patent Section, Dayton Office Staff* 43

Technical Presentations by GM Engineers 47

OTHER FEATURES

A Typical Problem in Engineering: Determine the Basic Design Specifications for a Ball Bearing Screw Assembly, *by George A. Widmoyer, Saginaw Steering Gear Division (assisted by Merle L. DeMoss, General Motors Institute)* 49

Solution to the Previous Problem: Determine the Interference Fit and Resulting Stresses in the Design of a Cold Extrusion Die, *by Charles R. Bradlee, Process Development Section (assisted by Robert W. Bund, General Motors Institute)* 51

Contributors to September-October 1955 issue of GENERAL MOTORS ENGINEERING JOURNAL 53

GENERAL MOTORS
Engineering
JOURNAL

Vol. 2 September-October 1955 No. 5

Published with the help of General Motors engineers everywhere

The GENERAL MOTORS ENGINEERING JOURNAL is published every other month by the Educational Relations Section of the Department of Public Relations, General Motors Corporation, and is intended primarily as a medium for presenting to the field of engineering education the results of General Motors research and engineering developments both in the product and production fields.

ADVISORY BOARD

C. A. CHAYNE, Vice President in Charge of Engineering Staff
J. J. CRONIN, Vice President in Charge of Manufacturing Staff
PAUL GARRETT, Vice President in Charge of Public Relations Staff
J. F. GORDON, Vice President and Group Executive
C. L. McCUEN, Vice President and General Manager, Research Laboratories Division
C. R. OSBORN, Vice President and Group Executive
S. E. SKINNER, Vice President and Group Executive
Secretary: K. A. MEADE, Director, Educational Relations Section

•

ARVID F. JOUPPI, Staff Assistant, Educational Relations Section
JOHN BRYANT, Editor
RAYMOND O. DARLING, Associate Editor
THOMAS F. MACAN, Assistant Editor
ANTON BEARD, Editorial Assistant
PATRICIA CLARK, Editorial Assistant

—

ERNEST W. SCANES, Art Director
JOHN B. TABB, Asst. Art Director

—

D. O. BRYSON, Production Manager

•

Address: EDUCATIONAL RELATIONS SECTION, Department of Public Relations, GM Technical Center, P.O. Box 177, North End Station, Detroit 2, Michigan.

•

Copyright 1955 by General Motors Corporation. All rights reserved under the International and Pan-American Copyright Conventions. Requests for permission to reproduce any part or parts of this publication may be directed to Joseph Karshner.

A Discussion of Economic Factors Affecting the Steel Selection and Heat Treatment for Automotive Gears

By VERNON E. HENSE and
DONALD P. BUSWELL
Buick Motor Division

Which gear steel and heat treating method should be used for volume production, high quality?

The economical manufacture of automotive rear axle, transmission, and flywheel ring gears is a function of four major factors—design, required quality, material selection, and heat treating process. The engineering design of an automotive gear establishes specific criteria pertaining to its desired quality. The selection of a steel and the type of subsequent heat treatment to satisfy the design and quality requirements necessitate a critical analysis by the engineer of many factors which have a governing influence on the economical aspects of the gear's manufacture. A knowledge of what requirements a specific steel is able to fulfill and what can be expected from the various types of available heat treating processes is essential. The specific steel and type of heat treatment selected are those which the engineer believes possess the optimum compromise between the qualities required for the particular gear application and overall cost.

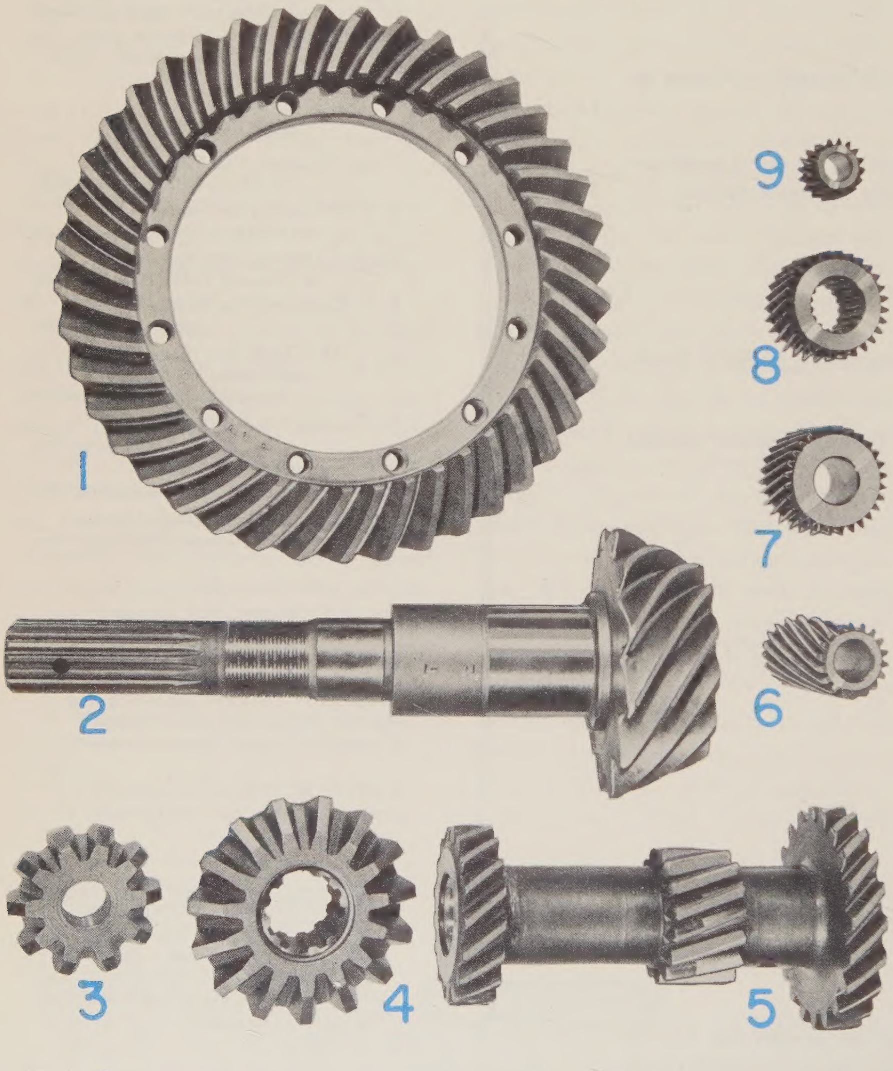


Fig. 1—These rear axle and transmission gears are produced in volume at Buick Motor Division and the steel and heat treating specification for each is determined by many factors. Rear axle parts shown are **1** ring gear, **2** drive pinion, **3** side pinion, and **4** side gear. Part **5** is a synchromesh transmission counter gear. Dynaflow automatic transmission parts shown are: **6** low planet pinion, **7** reverse planet pinion, **8** reverse sun gear, and **9** converter planet pinion.

AN OBJECTIVE of any industrial organization engaged in manufacturing a product for sale in a highly competitive market is to produce a product of the highest possible quality at the lowest possible cost. When the product has a number of component parts, each part must be given careful consideration in regard to the economical aspects of its design and manufacture. If the product must meet critical standards of performance when used for its intended application, the engineer is faced with the challenging job of utilizing sound engineering design to insure that all quality requirements will be met and that the product will be produced economically. The economic aspects of a product's manufacture depend upon its design and intended application and can be influenced by a number of factors.

Automotive rear axle and transmission gears represent just a few of the many components of a final end product—the automobile (Fig. 1). In order to fulfill the objective of manufacturing a product which is sold in a competitive market, the gears must be manufactured economically while meeting critical standards of performance—one of which is quietness of operation. To achieve these performance standards in the most economical manner the engineer must consider four major factors—design, required quality, material selection, and type of heat treating process.

The design of the gear establishes definite requirements pertaining to the quality which must be maintained. The design also has an influence on the processing costs and must take into consideration provisions which will insure that the gear can be produced economically within the necessary inspection limits and at the required production rate. The proper selection of a steel and the correct method of heat treating processing of this material to meet the design specifications and the required quality are absolute necessities for true economy. Great care must be exercised by the engineer when selecting a steel which not only must have adequate machinability to meet gear-tooth finish requirements but also must have favorable heat treating properties which will allow the required dimensional uniformity of the gear to be maintained.

Steel Selection

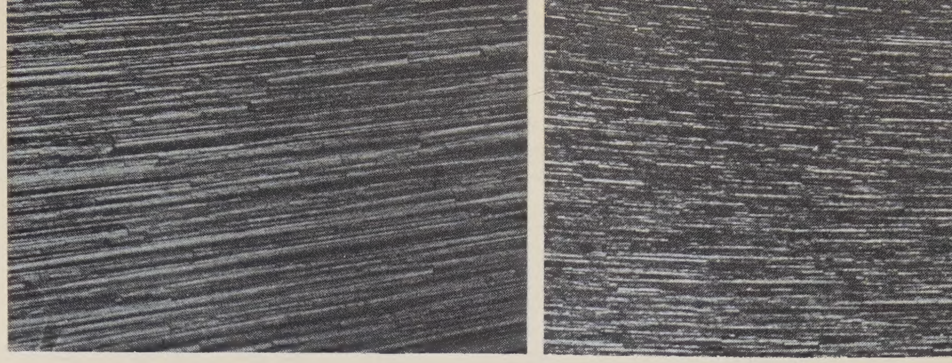
The requirements upon which the selection of a steel for an automotive gear application are based are dictated by the design, specific application, and required quality level of the gear. The proper selection can be made only after the engineer has considered definite factors which affect the overall economy of the gear's manufacture. Factors which the engineer must consider when selecting a steel are availability (of the material itself and the equipment to process it), cost, machinability and heat treating characteristics, and performance and uniformity properties.

Availability

There are many grades of steel available which afford the engineer a wide range of mechanical, physical, and metallurgical properties to choose from for an automotive gear application. Once a specific steel is selected, however, it must be available in sufficient quantities to fulfill production requirements. Conditions might arise where critical requirements, caused by design or quality demands, make it necessary to restrict the steel specification to a specialized grade. If a specialized grade of steel is selected and it is one which generally is not widely used, difficulty may be experienced in securing required quantities with suitable delivery.

Cost

The base cost of a steel, naturally, is of importance to the engineer when comparing one grade of a steel with equivalent grades. Any increase in the base cost of one grade of steel over another must be warranted by a necessary quality improvement or greater economy in processing. Because of the many intangibles involved, it is often quite difficult to use decreased processing costs as a factor in steel selection. Items of processing costs such as tools, repairs, and scrap can be evaluated only if adequate records are available for long production runs.



UNETCHED X-10

ETCHED X-10

Fig. 2—Many factors influence the surface finish of rear axle drive pinions. Shown above are macrophotographs of a rough (left) and smooth pinion. Both pinions were made of S.A.E. 4118 modified steel.

Machinability

The machinability of a steel generally is measured in terms of resulting surface finish and tool life—each of which can be a controlling factor. The engineer must select a steel whose composition will allow it to be machined easily to meet quality requirements at the specified rate of production and at a reasonable tool cost. Any differences in the inherent machinability between one grade of steel and another grade must be considered carefully before a final selection is made.

While the metallurgical information on a specific steel's machinability characteristics is highly important to the engineer when making a selection, the engineer must also consider other factors which have a bearing on the economic aspects of the gear's manufacture. For example, if the engineer selects a steel which is low in base cost but does not have the desirable machinability characteristics and if available heat treating equipment is inadequate to produce a structure which will give the required machinability, the base cost of the steel must be low enough to offset the purchasing of adequate heat treating equipment. Also, if the engineer selects a steel having machinability characteristics which can be improved by the addition of sulphur, the cost of the sulphur addition must be economically justified. The sulphur addition must have a desirable effect on tooth surface finish and must allow the production of gears with greater tooth accuracy and less stress from machining operations. Generally, steels which have sulphur additions are used only where low impact resistance is encountered and, therefore, are not used extensively for gears that are not in constant mesh, where clashing may occur.

Heat Treating Characteristics

When considering a steel's heat treating characteristics, the engineer must be careful to select a steel which will allow the specified gear design properties, such as required hardness, to be economically feasible with available heat treating equipment. If the demands of a gear's operation or its processing are such that the full variation of a steel specification will not produce satisfactory heat treating conditions, the engineer must either restrict the acceptable limits at an increased cost or be able to divert extreme heats of the steel to other gear applications.

With some automotive gear designs the engineer may experience difficulty in securing a steel which will give satisfactory heat treating results at the maximum and minimum extremes of a hardenability specification. Distortion, which causes erratic gear-tooth changes and subsequent increased straightening costs, is common at the high side of hardenability specifications, while unsatisfactory response in securing necessary properties may occur at the low side of hardenability specifications.

Performance Properties

The selection of a steel for an automotive gear application to meet required service characteristics has an effect on economy and must be given careful consideration by the engineer. Such characteristics as slight differences in wear resistance, toughness, or uniformity between various grades of steel become important when the design requirements of the gear and its specific application are critical. Experience, however, has shown that steels which prove unsatisfactory for automotive gear applications generally fail in heat treating and processing requirements, rather than in service requirements.

Uniformity of Properties

Automotive gears are designed for interchangeability and high-speed volume manufacture. It is highly important, therefore, that the engineer select a steel which has a high degree of uniformity. Any improvement in the uniformity of properties of one grade of steel over another cannot be overlooked as a factor influencing economy.

Selection of Heat Treating Process

Heat treatment is an operation or a series of operations involving the heating and cooling of a metal or alloy in the solid state for obtaining certain desired conditions or properties. The engineer may choose from many types of heat treating operations or processes to meet the design specifications for an automotive gear. Whether it be an annealing operation or a hardening operation, two general factors affecting economy must be considered:

- Cost of the heat treating operation.
- Effect of the heat treating operation upon a gear's subsequent processing.

The economy of the heat treating operation is a function of equipment, labor, operating cost, and maintenance cost.

The engineer selects heat treating equipment primarily to produce the required results demanded by the gear design specifications at the specified production rate. Sound judgment must be exercised when selecting heat treating equipment if an economical selection is to be made. For example, the cost of fuel in the immediate geographical area of production may determine whether the heat treating equipment will be heated by oil, gas, or electricity. The cost of fixtures, refractories, and miscellaneous maintenance items will be determined somewhat by the heat treating temperatures involved. The size of the equipment will depend upon the production rate and the flexibility of operation desired.

During the time that a gear is subjected to a heat treating operation its dimensions and properties are changed. Because of this action the engineer must thoroughly study the effect that a specific heat treating operation will have on a gear's subsequent processing. The changes in a gear's dimensions, referred to as *distortion* or *movement*, must be uniform in nature in order that these changes can be compensated for in machining. Any non-uniformity caused by a heat treating operation adds up to increased cost resulting from straightening, repairs, or scrap.

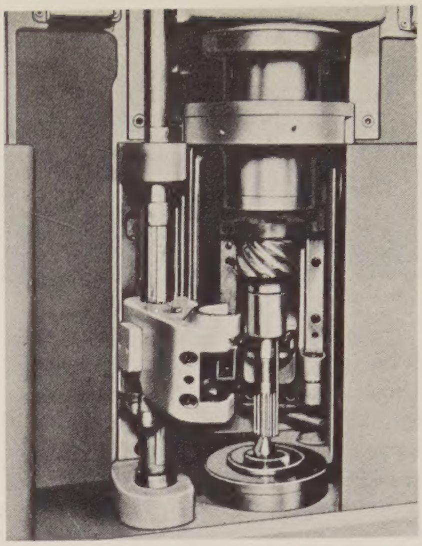


Fig. 3—The drive pinions for rear axle drive units are quenched in oil after they have been gas carburized. The operation is performed by a quenching machine. This view shows the machine door opened and the pinion held in the quenching position. After the carrier arm pivots back, the door closes, and a spinning cycle begins. Oil then flows around the pinion to complete the quenching operation. Rollers contact the pinion at its small bearing area to control distortion resulting from this operation.

The extent to which forging grain flow and annealed structure influence distortion during the heat treating process is not precisely known. However, good practice dictates keeping the forging grain flow as uniform as possible and concentric with the gear axis. In the same way, distortion caused by stresses set up during machining operations cannot be evaluated easily. Ideally, gears should be free from stress prior to the hardening operation. The stresses during the machining operations can be lessened by the use of a steel having a free-cutting structure.

Annealing

Annealing is a type of heat treating process which involves the heating and cooling of metals. Many types of annealing operations have been developed which make it possible to remove a metal's residual stresses; soften or refine a metal's grain structure; and alter ductility, toughness, or other physical properties of a metal (Fig. 2). The temperature of the specific type of annealing operation and the rate of cooling depend upon the particular steel being annealed and the desired effects.

The engineer applies an annealing heat treating process to automotive gears to produce a microstructure of satisfactory machinability and to relieve the material's residual stresses which might cause distortion in subsequent machining operations. The economy of annealing is a function of labor, operating and maintenance costs, and the effect of the annealing process on cleaning operations.

The particular type of annealing operation which the engineer chooses must be flexible enough to allow for changes to be made for steel variation. Three general methods are in use for annealing automotive gears. They are cycle annealing, normalizing, and isothermal transformation from the forging temperature.

In general, *cycle annealing* is an annealing process in which metal parts are heated to above the critical temperature, followed by controlled cooling to a suitable transformation temperature and held at that temperature to produce the required hardness and microstructure. *Normalizing* is a heat treatment in which iron-base alloys are heated to an appropriate temperature over the critical temperature. This is followed by cooling in air at room temperature. Steel parts can be isothermally transformed directly from the forging operation or from a suitable reheat temperature as with cycle annealing.

From the standpoint of cost, cycle annealing and normalizing annealing practices are about equivalent. A higher head temperature may tend to increase maintenance costs but, generally, slight variations in economy of operation are unimportant if improved results in machining or subsequent heat treatment can be realized. A normalizing furnace or a cycle annealing furnace can be operated with the same labor cost.

The isothermal transformation annealing process offers the engineer a variation in processing in which there are possible savings. In this particular annealing process, heat from a forging operation is used which eliminates handling and at the same time secures a structure which has good machinability characteristics.

The transformation process takes place in a salt bath at a range of approximately 1,100° F to 1,200° F. Another feature of this type of annealing process is that descaling, after annealing, is made simple because a flash water quench from the transformation temperature loosens scale formations. These savings, however, must be weighed against the cost of operation of the salt bath.

Hardening

Hardening is a term generally understood to imply increasing the hardness of a metal by an appropriate heat treating process. Examples of such heat treating processes include through hardening, carburizing, flame hardening, induction hardening, and carbonitriding.

The engineer applies a hardening process to automotive gears to produce the required physical properties with the least dimensional distortion. Distortion may occur because of stresses caused by machining operations, non-uniform heating and casehardening, or variations in a steel's hardenability characteristics.

When selecting a specific type of heat treating process for producing hardness in a gear it is essential that the engineer have a working knowledge of each process in order to select the proper one which will give the desired results at the most economical cost. Many factors must be taken into consideration if quality and uniformity are to be obtained

Two heat treating processes have contributed significantly to obtaining desirable gear hardness—gas carburizing and carbonitriding. *Gas carburizing* is a process which consists of circulating carbon-rich gas—such as city gas, natural gas, or various proprietary gas mixtures—around the gears. The entire operation may be carried out in continuous or batch-type gas-carburizing furnaces. Generally, the gas-carburizing operation is followed by direct quenching to produce the desired case and core properties in gears. *Carbonitriding* is a process of casehardening gears by the simultaneous absorption of carbon and nitrogen, brought about by heating gears in a gaseous atmosphere of suitable composition. The absorption process is followed by quenching in suitable media, depending on the requirements.

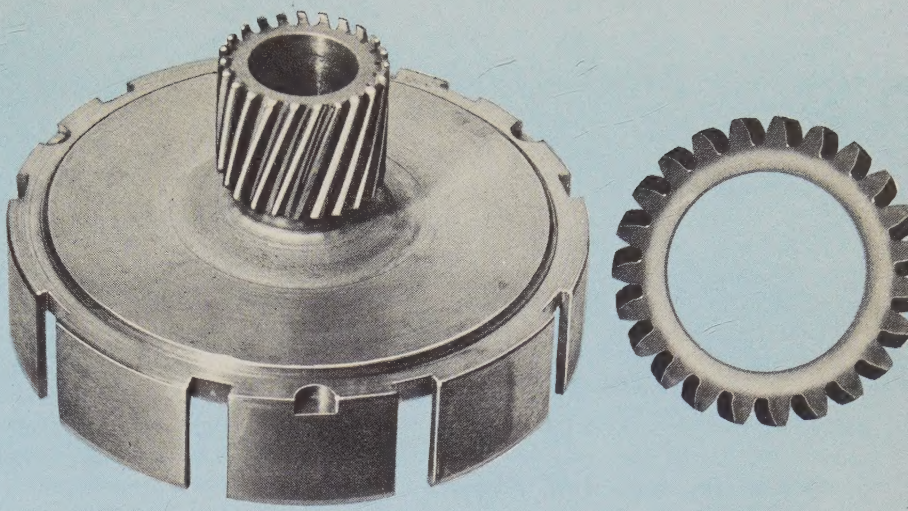


Fig. 4—The low-range reaction gear of the Dynaflow automatic transmission (left) is induction harden and quenched in oil. The hardening process produces a hardness pattern which extends from 0.025 in. 0.050 in. below the root of the tooth (right).

The gas carburizing and carbonitriding types of heat treating process offer the engineer specific advantages, such as uniformity of heating and quenching, flexibility in controlling case compositions, and high production.

Pack carburizing, sometimes referred to as box carburizing, is also used to obtain desirable automotive gear hardness. The gears to be treated are placed in a metallic container having a suitable carburizing compound. The containers are closed with a tight-fitting lid before being placed into a carburizing furnace in order to conserve the carburizing compound, to exclude circulating air, and to assist in retaining the carburizing atmosphere around the gears.

Quenching Operations

An important phase of heat treating hardening process is the quenching operation. The common quenching methods are immersion, die, and spray. There are also many quenching media available for the engineer's consideration. These include oil, water, air, and molten salt.

Experience has shown that quenching a dense load of gears in circulating oils at temperatures ranging from 100° F to 130° F makes it difficult to hold distortion to specified tolerances. Excessive distortion of gear teeth and shaft dimensions along with non-uniform changes in inside diameter dimensions are of major concern to the engineer if gear uniformity is to be realized. To minimize the variations in dimensions attendant with conventional oil quenching, die or fixture quenching may be used, or the media may be hot oil (300° F to 400° F) or molten salt.

When excessive distortion is encountered it often is economical to incorporate a hot quenching operation using either oil or salt as the quenching medium. *Hot quenching* is a method whereby quenching of iron-base alloys takes place in a medium, the temperature of which is substantially higher than room temperature. Hot quenching cycles, although serving to minimize distortion substantially, do not eliminate it entirely. Some hot quenching operations are done in continuous furnaces where the quenching unit is conveyorized. In other hot quenching operations, the gears are heated in batch-type furnaces and batch quenched.

Since hot quenching treatments, at the present time, are generally more costly than the more conventional quenching treatments the engineer must balance the improvement in uniformity achieved against the increased expense. Unless considerable scrap or rework is eliminated, it often is impossible to justify a hot quenching treatment.

Steel and Heat Treating Process Selection in the Automotive Industry

From the foregoing, it may be deduced that each manufacturing organization might select different steels and different related heat treating processes. To substantiate this deduction, during 1955 a survey was made of current metallurgical practices in several automotive manufacturing organizations. The survey sought only to determine practice, without regard for economic and other factors, and was limited to information on steels for rear axle, transmission, and flywheel ring gears. In the transmission group,

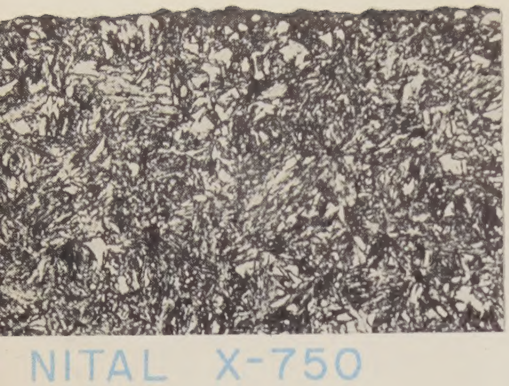
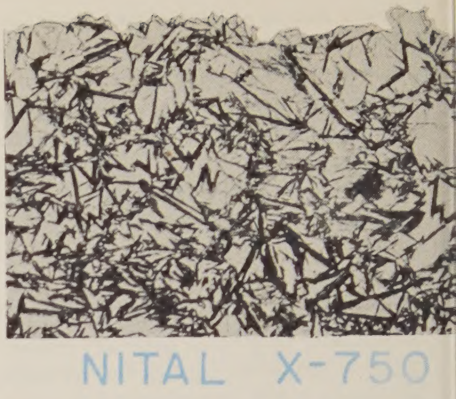


Fig. 5—Conventional transmission gears of S.A.E. 1340H steel were first gas carburized with an ammonia concentration of 0.5 per cent of the carburizing atmosphere. This amount of ammonia resulted in the microstructure shown on the left. Then the ammonia concentration was increased to 1.5 per cent of the atmosphere. The resulting microstructure is shown on the right.



gears for the conventional, semiautomatic, and automatic types were included. The results of the survey are contained in the appendix to this paper.

(A similar survey was made in 1949 and 1950 and the results presented in a paper "Economics of Automotive Gear Steels and Their Heat Treatment," by V. E. Hense, H. H. Miller, and R. B. Schenck, Buick Motor Division, and delivered at the 1950 Summer Meeting of the Society of Automotive Engineers in French Lick, Indiana.)

Selection in Buick Motor Division

A number of examples of Buick Motor Division practice are of interest in explaining in specific terms how these technical and economic factors are considered in a typical automobile manufacturing operation. In recent years, Buick Motor's engineers have selected low-alloy steels for automotive gear application. The introduction of low-alloy steels for automotive gears required a great deal of testing in order to secure adequate cost figures for comparison with the steels previously used. The test results showed that the low-alloy steels represented a decrease in cost which was great enough to absorb any slight increases in processing costs necessary with the low-alloy steels.

It is difficult to justify a change in steels solely because of slight advantages in cost of one material over another. In the application of new steels it is usually possible to secure the required quality but generally is somewhat difficult to fit the new material into existing equipment to secure the required machinability, and to control heat treating distortion.

Rear Axle Gears

The Buick rear axle drive unit consists of a ring gear, side gear, side pinion, and drive pinion (Fig. 1). Experience gained by Buick Motor's engineers showed that it is possible, with the same gear design, to use different types of steel successfully for the rear axle gears and pinions. At the present time, S.A.E. 4118 steel is being used for rear axle ring gears. The side gears are made of S.A.E. 1132 steel, the side pinions of S.A.E. 1016 steel, and S.A.E. 4118 modified steel is used for the drive pinions. The modification increases the manganese range from 0.70-0.90 per cent to 0.70-1.00 per cent and the chromium range from 0.40-0.60 per cent to 0.50-0.75 per cent.

In the annealing process, the important consideration is the development of a microstructure which will furnish the required smoothness and high-quality surface finish for gear teeth. This requirement is met on rear axle ring gears and drive pinions by an annealing treatment. The cycle used consists of heating to 1,750° F, rapid cooling to 1,200° F, and holding at 1,200° F until transformation is complete. The hardness of the ring gears and drive pinions is generally between 143 Bhn to 163 Bhn. Fig. 2 shows extremes in gear-tooth surface finish on a drive pinion made of S.A.E. 4118 modified steel.

In order to control the quality of gears coming from the heat treating operations, Buick Motor's production departments run *pilot lots* for each new heat of steel in order to determine the distortion characteristics. Daily checks also are made to control the quality of gears coming from the heat treatment. Any changes in tooth cutting to allow for tooth distortion are made only when definite quality trends have been established. The variation in hardenability from one heat of steel to another makes quality control a necessity. The narrower the hardenability spread of the steel the better chance to produce uniform gears in the heat treating operations. Any narrowing of hardenability bands is a step toward more desirable production conditions and decreased manufacturing costs. A full spread of hardenability for an available carburizing steel is too great for the best heat treating conditions.

Ring gears, side gears, and drive pinions are gas carburized in a continuous furnace to a case depth of from 0.40 in. to 0.50 in. The carburizing temperature is 1,650° F.

To maintain dimensional accuracy during the quenching operation, the ring gears are die-quenched in oil. To eliminate straightening operations after hardening, specially designed quenching machines are used to quench the drive pinions directly into circulating oil after the carburizing operation (Fig. 3). Ring gears and drive pinions are tempered at 350° F. The side pinions are quenched directly into water from the carburizing operation.

The side gears of the rear axle drive unit at one time were made from S.A.E. 1016 steel. The gears were pack carburized to a case depth of from 0.040 in. to 0.050 in., slow cooled, reheated, and finally quenched in water. Recurring difficulty with erratic distortion caused Buick Motor's engineers to change the steel specification; presently used steel is S.A.E. 1132. The gears are now carbonitrided to a case depth of from 0.015 in. to 0.025 in. and oil quenched in a continuous furnace. The change in steel specification and type of heat treating process eliminated the previously encountered difficulty with distortion.

Experience has shown that a major factor influencing distortion is the carbon content of the carburized case. The maximum carbon content of the case has an effect upon the hardenability which, in turn, influences dimensional changes during the heat treating process. This variation in hardenability, as affected by the maximum carbon content, changes from one steel to another and in extreme cases the specified hardness cannot be attained. Buick Motor's engineers have found that a surface content of from 0.85 per cent to 0.95 per cent carbon is desirable. A more flexible control of case carbon content is possible with gas carburizing, although satisfactory control also can be had with pack carburizing by the careful mixture of new and used carburizing compounds.

Dynaflow Torque-Converter Automatic Transmission Gears

In order to maintain tooth accuracy and uniformity required for automatic transmission gears, selective hardening of gear teeth and hot quenching treatments are used extensively.

Dynaflow automatic transmission pinions are made from S.A.E. 1330H steel carburized to 0.007-0.012 in. in either batch-type or continuous furnaces, quenched in hot oil, and tempered at 400° F.

When this transmission was introduced, the low-range reaction gear, for example, was made of S.A.E. 1050 steel. The gear was induction hardened to a depth of 0.050 in. below the root of the gear teeth. During the induction hardening process, a water quench was used. The water was directed only on the inside of the gear hole and was not allowed to make contact with the gear teeth. This practice was continued until low hardenability heats of steel were encountered which would not harden satisfactorily. An arrangement was then made whereby the quench water was allowed to make direct contact with the gear teeth. This practice, however, caused some quenching cracks. These difficulties led to the presently used oil quenching medium and S.A.E. 1052 steel in order to provide sufficient hardenability. Fig. 4 shows the presently used low-range reaction gear and tooth hardness pattern.

Conventional Transmission Gears

The problem which confronts the engineer in annealing gears for a conventional transmission is to secure a microstructure which will satisfy the varied machining operations at an acceptable tool cost. The machining operations performed on transmission gears include broaching, turning, milling, drilling, hobbing, and shaving.

Occasionally, design or finish requirements for a gear are such that broaching becomes a critical operation. This requires an annealing operation that results in a microstructure favorable to broaching and increased hardness. As a result, the machinability is somewhat decreased for the turning and hobbing operations. At the present time, S.A.E. 1340H steel also is being used as the steel specification for Buick conventional transmission gears.

Continuous gas carburizing furnaces were first used for the heat treatment of conventional transmission gears. Initially, ammonia was not added to the carburizing atmosphere. When it was decided later to use ammonia, the ammonia addition amounted to 0.5 per cent of the atmosphere and difficulty was encountered with erratic casehardening. Experience with this situation pointed out that, to reduce the tendency toward gear-tooth pitting, either a greater case depth or more of a nitrogen effect was necessary. It was decided finally to leave the depth of the case unchanged and increase the ammonia addition up to the presently used 1.5 per cent of the total atmosphere. This resulted in a more favorable case structure at the gear surface and did not require the purchase of additional heat treating equipment to increase the case depth (Fig. 5). Now conventional transmission gears are carbonitrided in continuous furnaces to a case depth of from 0.005 in. to 0.010 in., with the usual practice being to hold the case depth to the high side of the specification.

The quenching of transmission gears offers the engineer the best opportunity for decreasing the manufacturing cost. As a result, hot quenching methods are used which improve the control of dimensions and reduce the amount of straightening necessary.

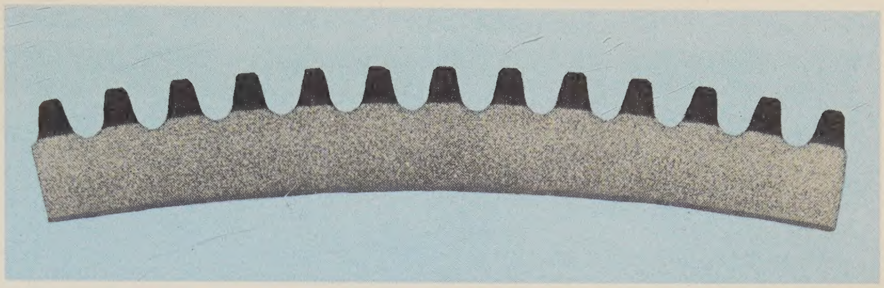


Fig. 6—The Buick flywheel ring gear is made of S.A.E. 1045 steel and the teeth are flame hardened. The tempering operation is carried out at the same time that the gear is shrunk on the flywheel.

Flywheel Ring Gears

The flywheel ring gear in the Buick automobile is made from S.A.E. 1045 steel and the gear teeth are flame hardened. Quenching is accomplished by immersion in oil. Tempering, by means of an induction heater, is carried out simultaneously with the operation of shrinking the gear on the flywheel.

At one time, Buick cars had a flywheel ring gear with a very light carburized case. This ring was surface treated in a gas carburizing furnace. The gears were hardened by oil quenching and the tempering operation was separate from the heating operation for shrinking the gear on the flywheel. With the introduction of the Dynaflow automatic transmission, the cast iron flywheel was made obsolete. It then became necessary to get a more accurate gear than was required when the gear was previously shrunk on the rigid cast-iron flywheel. In view of this, the change was made to selective hardening. Fig. 6 shows a tooth hardness pattern obtained by the present flame heating operation on the flywheel ring gear.

Summary

The economical manufacture of high quality automotive gears requires the consideration of many factors by the engineer. The specific design and operating requirements which each gear must meet necessitate a thorough analysis of the type of steel to be used and the type of heat treating process to which the steel will be subjected. The engineer must base the steel selection upon the gear's design, specific application, and required quality level. He also must consider such items as the steel's cost, availability, machinability, heat treating characteristics, and performance and uniformity properties. The selection of a specific type of heat treating process—whether it be an annealing or hardening operation—is based upon the cost of the heat treating operation and its overall effect upon the gear's subsequent processing. Many types of steels and annealing and hardening processes are available to the engineer from which to make a selection. The final selection which is made must be one the engineer believes will satisfy all design requirements and fulfill the overall objective of competitive manufacture—produce a product of the highest possible quality at the lowest possible cost.

Appendix

The preceding basic paper has discussed the economic factors affecting the steel selection and heat treatment for automotive gears, and has applied these factors to Buick Motor Division's technical decisions which have resulted in overall satisfactory production results. These metallurgical decisions were affected by the whole of Buick Motor's engineering and manufacturing skills and, as importantly, its facilities. Facilities, if not skills, may vary from one plant to another and Buick Motor's solutions may not be the right ones for other manufacturing operations. The comparison of current practice in representative operations, including Buick Motor, contained in Tables I through VIII shows what each manufacturer has found to best satisfy requirements and, as such, must be assumed as representing economy in the selection of steels and heat treatments. They are presented here for their educational value, and the authors wish to acknowledge the cooperation of the men and companies which contributed information.

S.A.E. REAR AXLE GEAR AND PINION STEEL SELECTION

Manufacturer	Drive Pinion	Ring Gear	Side Gear	Side Pinion
A	8617	8615	5130	5130
B	4620	4620	1024	1022
C	8617H	8617H	8617H	8617H
D	8620	8620	8615	8615
E	8620	8620	1024	1024
F	1024	1024	1024	1024
G	4028	4028	8620H	8620H
H	4028 8620*	4028	4027	4024
I	4118 Mod.**	4118	1132	1016

*Large drive pinion

**Carbon 0.17-0.23, Manganese 0.70-1.00, Chromium 0.50-0.70, Molybdenum 0.08-0.15

Table I—Present practices of those automotive manufacturers surveyed shows a wide variety in the selection of S.A.E. steels for rear axle gears and pinions.

S.A.E. MANUAL TRANSMISSION GEAR STEEL SELECTION

Manufacturer	Counter Gear	Clutch Gear	Low Speed Gear	2nd Speed Gear	Idler Gear
A	8620	8620	8620	8620	8620
B	5117	5117	5117	5117	1340H
C	8640H	8640H*	8640H	8640H	8640H
D	8620H	8620H	8620H	8620H	8620H
F	1024	1024	1024**	1024	1024***
G	5135H	5135H#	5135H	5135H##	5135H
H	4027	4027###	4027	4027	4027
I	1340H	1340H	1340H	1340H	1340H

*Clutch shaft

**Selected heats to maintain 25-35 RC core

***Reversing pinion

#Main drive gear

##Intermediate gear

###Transmission drive pinion

Table II—Present practice of those automotive manufacturers surveyed shows a variation in the S.A.E. steel designation for manual transmission gear steels.

REAR AXLE GEAR STEEL ANNEALING PRACTICES

MANUFACTURER	DRIVE PINION		RING GEAR		SIDE GEAR	
	S.A.E. STEEL	TREATMENT PRIOR TO MACHINING	S.A.E. STEEL	TREATMENT PRIOR TO MACHINING	S.A.E. STEEL	TREATMENT PRIOR TO MACHINING
A	8617	Anneal—1775-1200° F	8615	Anneal—1775-1200° F	5130	Normalize—1650° F
B	4620	Anneal—1750° F Fast cool to 1200° F Slow cool to 600° F	4620	Anneal—1750° F Fast cool to 1200° F Slow cool to 600° F	1024	Anneal—1750° F Fast cool to 1200° F Slow cool to 600° F
C	8617H	Anneal—1750-1100° F	8617H	Anneal—1750-1100° F	8617H	Anneal—1750-1100° F
D	8620	Anneal—1700-1250° F	8620	Anneal—1700-1250° F	8615	Anneal—1700-1250° F
E	8620	Anneal—1825-1150-1000° F	8620	Anneal—1825-1150-1000° F	1024	Bar stock
F	1024	Normalize—1600° F	1024	Normalize—1600° F	1024	Normalize—1600° F
G	4028	Anneal— Forging heat to 1190° F salt	4028	Anneal— Forging heat to 1190° F salt	8620H	Anneal— Forging heat to 1190° F salt
H	4028 8620*	Anneal—1750° F Air blast cool to 1000° F in 10 min	4028	Anneal—1750° F Air blast cool to 1000° F in 10 min	4027	Anneal—1750° F Air blast cool to 1000° F in 10 min
I	4118 Mod.	Anneal—1750° F-1200° F	4118	Anneal—1750° F-1200° F	1132	Anneal— 1600-1130-1100-1075° F

*Large drive pinion

Table III—The survey shows that annealing practices for rear axle gear steels vary according to metallurgical selections, available processes, and desired quality.

REAR AXLE GEAR STEEL HEAT TREATING PRACTICES

Manufacturer	Part Name	S.A.E. Steel	Method	Case Depth (in.)	Quench Medium		Tempering Temp (° F)
					Type	Temp (° F)	
A	Drive Pinion	8617	Gas Carburize	0.045-0.060	Oil	130-140	325
	Ring Gear	8615	Pack Carburize	0.040-0.055	Oil	130-140	325
	Side Gear	5130	Gas Carburize	0.005-0.015	Oil	130-140	350
	Side Pinion	5130	Gas Carburize	0.005-0.015	Oil	130-140	350
B	Drive Pinion	4620	Gas Carburize	0.045-0.055	Oil	115	400
	Ring Gear	4620	Gas Carburize	0.045-0.055	Oil	115	375
	Side Gear	1024	Gas Carburize	0.045-0.055	Oil	115	350
	Side Pinion	1022	Pack Carburize	0.045-0.055	Caustic	85	350
C	Drive Pinion	8617H	Gas Carburize	0.040-0.045	Oil	115	None
	Ring Gear	8617H	Gas Carburize	0.040-0.045	Oil	115	325
	Side Gear	8617H	Gas Carburize	0.040-0.045	Oil	115	None
	Side Pinion	8617H	Gas Carburize	0.040-0.045	Oil	115	None
D	Drive Pinion	8620	Gas Carburize	0.040-0.050	Oil	130	None
	Ring Gear	8620	Gas Carburize	0.040-0.050	Oil	130	None
	Side Gear	8615	Gas Carburize	0.040-0.050	Oil	130	None
	Side Pinion	8615	Gas Carburize	0.040-0.050	Oil	130	None
E	Drive Pinion	8620	Gas Carburize	0.040-0.050	Oil	110	300
	Ring Gear	8620	Gas Carburize	0.040-0.050	Oil	110	425
	Side Gear	1024	Gas Carburize	0.040-0.050	Oil	110	300
	Side Pinion	1024	Gas Carburize	0.040-0.050	Oil	110	300
F	Drive Pinion	1024	Gas Carburize	0.030-0.040	Oil	130-140	350
	Ring Gear	1024	Gas Carburize	0.030-0.040	Oil	130-140	350
	Side Gear	1024	Gas Carburize	0.020-0.030	Oil	130-140	350
	Side Pinion	1024	Gas Carburize	0.020-0.030	Oil	130-140	350
G	Drive Pinion	4028	Gas Carburize	0.045-0.055	Oil	130-140	360
	Ring Gear	4028	Gas Carburize	0.040-0.050	Oil	130-140	360
	Side Gear	8620H	Gas Carburize	0.030-0.040	Oil	130-140	360
	Side Pinion	8620H	Gas Carburize	0.030-0.040	Oil	130-140	360
H	Drive Pinion	4028	Gas Carburize	0.040-0.050	Oil	100-120	350-375
		8620*	Gas Carburize	0.040-0.050	Oil	100-120	350-375
	Ring Gear	4028	Gas Carburize	0.040-0.050	Oil	100-120	350-375
	Side Gear	4027	Gas Carburize	0.030-0.040	Oil	100-120	350-375
	Side Pinion	4024	Gas Carburize	0.030-0.040	Oil	100-120	350-375
I	Drive Pinion	4118 Mod.	Gas Carburize	0.040-0.050	Oil	130-150	350
	Ring Gear	4118	Gas Carburize	0.040-0.050	Oil	130-150	350
	Side Gear	1132	Gas Carburize	0.015-0.025	Oil	130-150	400
	Side Pinion	1016	Gas Carburize	0.040-0.050	Water		None

*Large Drive Pinion

Table IV—Gas carburizing is widely practiced in heat treatment of rear axle gear steels. Note the rather wide variance on other metallurgical selections.

AUTOMATIC AND SEMIAUTOMATIC TRANSMISSION PINION HEAT TREATING PRACTICES

Manufacturer	S.A.E. Steel	Treatment Prior to Machining	Hardening Method	Case Depth (in.)	Quench Medium		Tempering Temp (° F)
					Type	Temp (° F)	
A	5145	Bar Stock	Gas Carburize	0.007-0.010	Oil	125-130	440
F	1024	Bar Stock*	Gas Carburize	0.020-0.030	Oil	130-140	350
H	4028	Bar Stock	Gas Carburize	0.030-0.040	Oil	170-180	400
I	1330H	Bar Stock	Gas Carburize	0.007-0.012	Oil	325	400
J	5140H	Bar Stock	Gas Carburize	0.005-0.010	Oil	300	375-510
	5145H	Bar Stock	Gas Carburize	0.005-0.010	Oil	300	510

*1200° F after rough machining

Table V—This chart shows automatic and semiautomatic transmission pinion practice of those automotive manufacturers surveyed.

MANUAL TRANSMISSION GEAR STEEL ANNEALING PRACTICES

Manufacturer	S.A.E. Steel	Treatment Prior to Machining	Temperature (° F)
A	8620	Normalize	1750
B	5117	Normalize	1700
	1340H*	Anneal	1600-1130-1100-1075
C	8640H	Anneal	1750-1200
D	8620H	Anneal	1700-1200
F**	1024	Normalize	1600
G	5135H	Forging Heat to	1260
		or	
		Anneal#	1600-1260
		Anneal (bar stock)##	1750-1000
H	4027	Anneal	1750-1600—Blast cool to 1050
I	1340H	Anneal	1600-1130-1100-1075

*Idler gear

**Second speed gear and reversing pinion not normalized

#Counter gear, main drive gear, and low and reverse sliding gear

##Intermediate gear and reverse idler gear

Table VI—As with rear axle gears (Table III), annealing practices for manual transmission gear steels vary according to metallurgical selections, available processing, and other factors.

MANUAL TRANSMISSION GEAR STEEL HEAT TREATING PRACTICES

Manufacturer	S.A.E. Steel	Method	Case Depth (in.)	Quench Medium		Tempering Temp (° F)
				Type	Temp (° F)	
A	8620	Gas Carburize	0.035-0.050	Oil	120-130	300-350
B	5117	Gas Carburize	0.040-0.050	Oil	110-130	300
	1340H*	Gas Carburize	0.005-0.010	Oil	130-150	400
C	8640H	Gas Carburize	0.008-0.015	Oil	115	425
D	8620H	Gas Carburize	0.040-0.050	Oil	180 or	375
				Salt	400	
F	1024	Gas Carburize	0.030-0.040**	Oil	130-140	350#
			0.020-0.030***	Oil	130-140	350
G	5135H	Gas Carburize	0.012-0.016	Oil	130-140	360
H	4027	Gas Carburize	0.030-0.040	Oil	145-175##	400-410
I	1340H	Gas Carburize	0.005-0.010	Oil	130-150	400

*Idler gear

**Low speed and 2nd speed gear

***Clutch gear and reversing pinion

#400° F on low speed gear

##175° F to 200° F on drive pinion

Table VII—All manufacturers surveyed use gas carburizing as their method for heat treatment of automotive manual transmission gear steels. The most commonly used quench medium is oil.

FLYWHEEL RING GEAR STEEL SELECTION AND HEAT TREATING PRACTICES

Manufacturer	S.A.E. Steel	Treatment Prior to Machining	Hardening			Tempering Temp (° F)
			Method	Quench Medium		
				Type	Temp (° F)	
A	1050 Mod.*	Normalize 1420° F	Furnace	Oil	130-140	750
C	1045	Anneal 1600° F	Flame	Oil	115	500-600
F	1050	Normalize 1400° F	Flame	Oil	130-150	500
G	1046	Normalize 1650° F	Induction	Water**		Induction—700
H	1045	Normalize 1800° F	Flame	Oil	100-120	Induction—500-600
I	1045	Normalize 1600° F	Flame	Oil	130-150	Induction—500-550
J	1045	—	Flame	Oil	130-150	550

*Carbon 0.48-0.53
Manganese 0.85-1.05 Coarse Grained

**Gear removed from water at about 550° F

Table VIII—Flywheel ring gear practice of those automotive manufacturers surveyed shows some uniformity in S.A.E. steel selection but attendant differences in processing.

Toroidal-Type Current Meter Improves Weld Quality

By PATON M. ZIMMERMAN
Fisher Body Division

New application

for an

old principle

Production of quality spot welds and projection welds requires proper control of the air or hydraulic pressure providing the electrode clamping force, the amount of welding current, and the time of current flow. Of these three variables regulated by the welding machine, the most important is the welding current. The welding current meter developed by Fisher Body Division's engineers provides a simple, convenient means of measuring actual welding current at the electrode tips of the welding machine. Previously available instrumentation of comparable accuracy was much more expensive and bulky and required the services of engineering personnel. Now used in production plants by welder maintenance men, the newly developed welding current meter has contributed greatly to the improvement of weld quality.

The strength, appearance, and durability of an automobile body are largely dependent on the quality of the resistance welding used to assemble the various body components. The two resistance welding processes most widely used by the Fisher Body Division are spotwelding and projection welding.

For consistently good resistance welding that is uniform in quality from one machine to another and between different plants, it is important that optimum resistance welding schedules be observed. A *welding schedule* is a listing of the proper welding force, welding current, weld time, and electrode tip shape for the correct joining of a specified combination of metal compositions and thicknesses.

In the past, the most difficult welding variable to measure has been welding current, particularly when measurements were desired under plant operating conditions. Due to the large currents encountered in resistance welding (6,000 amp to 24,000 amp), direct measurements of welding current are not practical. Even the lower current in the primary of the welding transformer (100 amp to 1,500 amp) still necessitates the use of a current transformer to provide measurable currents.

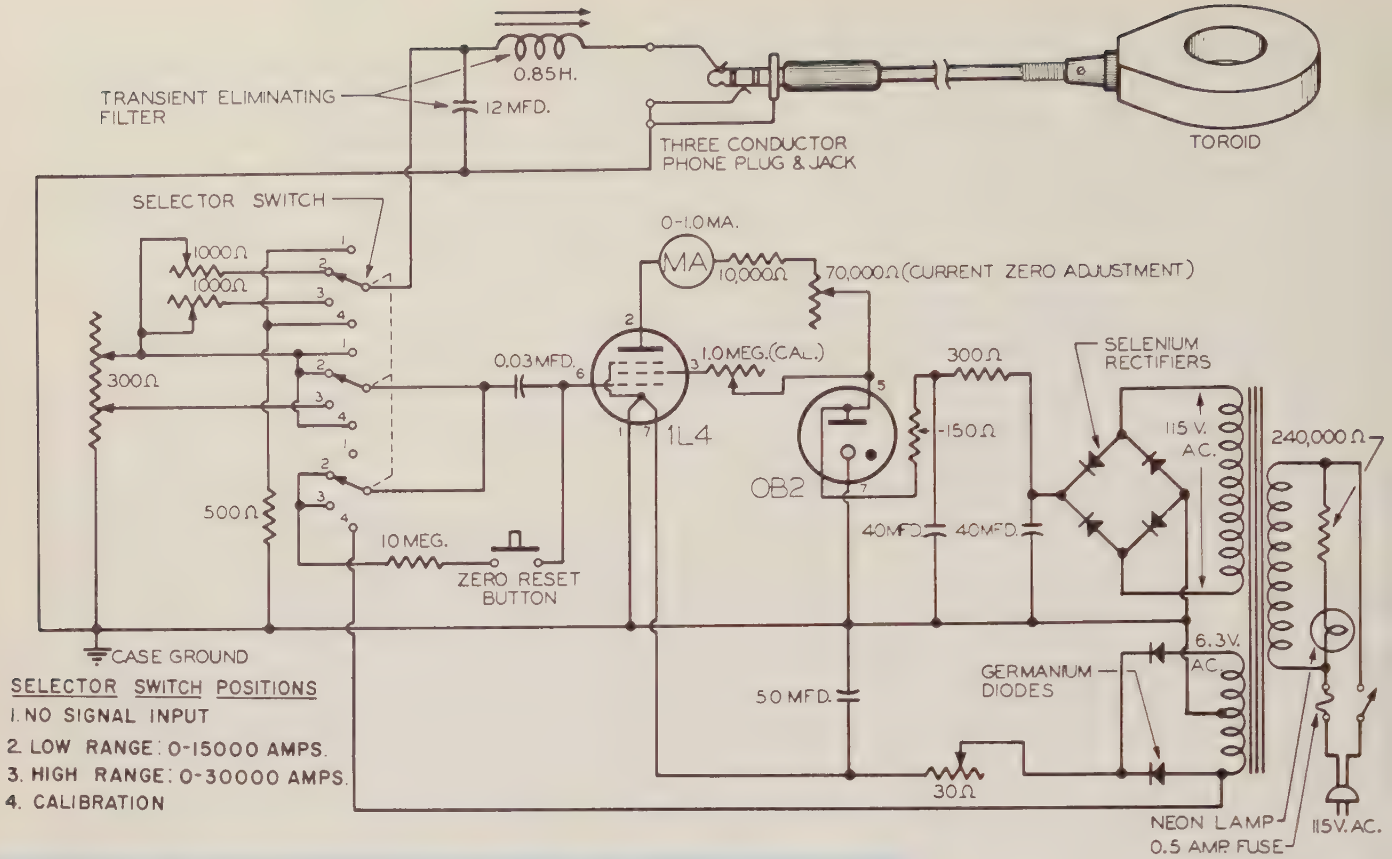
Several methods of measuring the current in the secondary welding circuit have been used. High resistance shunts have been built into the secondary circuit, usually in one of the electrode holders, to provide a measurable voltage drop. Iron-core current transformers have been placed around one of the current-carrying members of the secondary circuit or around one of the primary leads to the welding transformer. All of these instrumentation methods have posed problems such as, the need for frequent and careful calibration, the necessity for several current transformers to cover the range of currents encountered, the loading of the secondary circuit with magnetic materials thereby changing the circuit inductance, or the requirement of disconnecting and reconnecting the wiring of the welding equipment.

It was considered highly desirable to have an instrument for measuring welding current which would require no electrical connections to the welding equipment, have no effect on the characteristics of the welding circuit, and provide means for direct reading. In addition, a satisfactory instrument would have to be portable and receive its power from a standard 110-v supply line.

An air-core transformer or toroidal coil seemed to be the most logical solution to the problem as it contained no magnetic material to change the impedance of the welding circuit and could be placed over one of the welding electrodes. Since the voltage produced in a toroidal coil surrounding a current-carrying conductor is proportional to the rate of change of magnetic flux produced by the current, it was necessary to connect the toroid to a suitable measuring circuit that would permit a direct reading in terms of current.

The basic principle of the welding current meter consists of measuring the voltage produced by a toroidal (doughnut shaped) coil which is placed around a conductor carrying the current that is to be measured. The toroidal voltage is a function of the rate of change of magnetic flux produced by the current flowing through the conductor and the resistive load across the toroid. Only the magnetic flux produced by the current-carrying conductor, which is inside the toroid, will have any effect on the toroidal voltage; magnetic flux produced by conductors outside the toroidal loop, even though they may be immediately adjacent to the toroid, do not produce any toroidal voltage. The toroid is a differentiating unit which means that, if the conductor passing through the toroid is carrying a current with a sine wave form, the induced voltage in the toroid will cause a current to flow through the load across the toroid which has a cosine wave form and is 90° out-of-phase with the original current. Fortunately, the cosine wave form is identical in shape to the sine wave form, but if wave forms other than a sine wave are carried in the conductor, the differentiated toroidal output current will not resemble the original current in wave form.

In order to measure the output voltage of the toroid, a peak-reading, a-c, vacuum-tube voltmeter circuit is employed, since the welding currents last for such short periods of time that it would be impractical to use conventional a-c voltmeters. Therefore, the toroidal voltage is used to control the grid bias of a vacuum tube, thereby affecting the plate current which the tube will conduct. A condenser in the grid circuit is charged, by grid rectification of the vacuum tube, to the peak value of voltage developed by the toroid. This condenser charge or voltage determines the negative grid bias of the tube. The amount of plate current passing through the tube is proportional to the



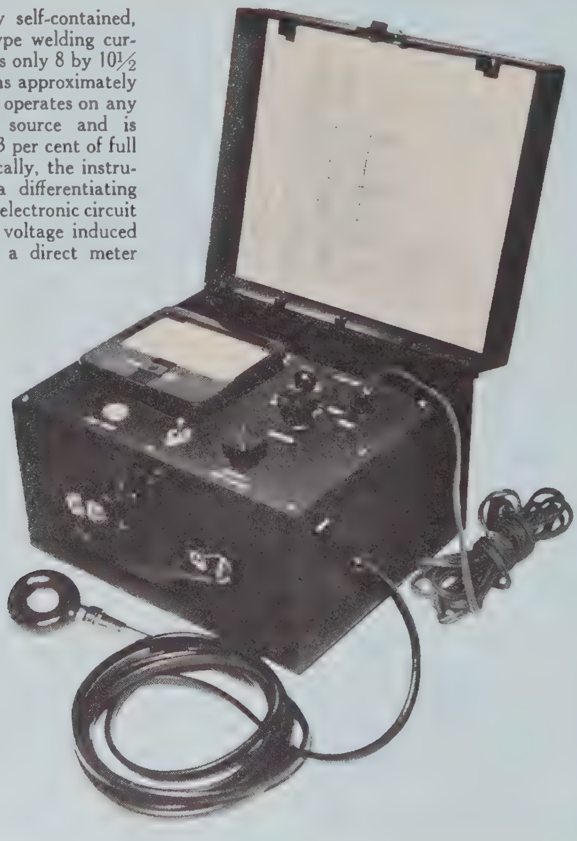


Fig. 1—Completely self-contained, the new toroidal-type welding current meter measures only 8 by 10½ by 11 in. and weighs approximately fourteen pounds. It operates on any 110-v, a-c power source and is accurate to within 3 per cent of full scale reading. Basically, the instrument consists of a differentiating toroidal coil and an electronic circuit which resolves the voltage induced in the toroid into a direct meter reading in amperes.

grid bias and is read by a milliammeter. The short impulse voltage generated by the toroid is translated by this method to a steady-state meter indication unaffected by the length of duration of the toroidal voltage. The circuit is designed so that the maximum plate current of 1.0 ma (milliampere) will flow when the toroidal output voltage is zero; any voltage produced by the toroid will increase the negative grid bias of the tube and decrease the amount of plate current. For example, if no welding current is flowing the meter needle will appear in the extreme right position, corresponding to maximum plate current. On the other hand, if maximum welding current that can be measured is flowing, zero plate current will result and the needle will appear at the extreme left of the meter. The meter readings in milliamperes are recalibrated as a straight-line function in terms of kiloamperes of secondary welding current, which results in a multiplication factor of one million to one. A variable resistance in series with the meter permits adjusting the maximum plate current to exactly 1.0 ma. A switch is provided to discharge the grid condenser after each reading so that the original grid bias is restored.

In addition to this basic instrument, it was felt that certain other features should be included in the development of such a measuring device. It was decided to provide an electronically regulated power supply. A calibration circuit to check proper operation of the instrument was considered essential as was a stand-by position to permit the instrument to be left in operating condition (warm-up takes about three minutes) while not actually indicating welding current. A serious limitation of previous designs from the standpoint of Fisher Body applications was that, although not influenced by transients caused by the initiation of the welding current by mechanical or electronic contactors, early experimental models were seriously affected by transients produced when the welding current was interrupted—particularly by arcing of mechanical contactors when the current was broken. This arcing is a very common occurrence, and the inability of the instrument to be used in such a situation would prevent its general use. Therefore, a satisfactory secondary current-measuring device had to be introduced in order to overcome this objection.

In the original design, as well as in the first experimental model, the toroid was composed of six individually wound coils of 2,000 turns each, arranged around the circumference of a circular form and connected in series. This arrangement was bulky, as well as expensive to manufacture. It was considered desirable, therefore, to utilize commercially available toroidally wound coils that were rugged enough to withstand the normal shop treatment of instruments.

In the final design of the welding current meter, the triode vacuum tube of a previous model was replaced with a tube of nearly similar characteristics (Fig. 1). The best equivalent proved to be a 1L4 pentode vacuum tube; the additional screen grid afforded a convenient method of controlling the gain of the tube by varying the screen grid voltage. A grid condenser value of 0.03 microfarads was selected as affording the most stable meter response.

The input voltage from the toroid is passed through an L-type, low-pass, filter network consisting of an inductance of 0.85 h (henries) shunted with a 12-microfarad condenser. This filter greatly attenuates frequencies above 60 cps (cycles per second) (cut-off frequency is 180 cps) preventing transients—which have a frequency higher than the 60-cps frequency of the welding current—from affecting the meter reading. This eliminates the breaking transients which consist primarily of high-frequency components.

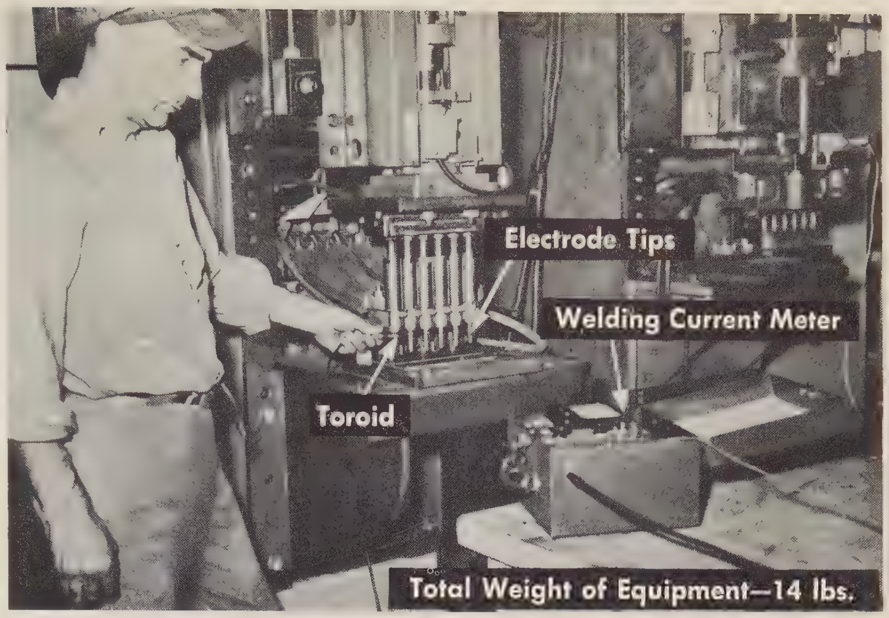


Fig. 2—The new toroidal-type welding current meter is a direct-reading instrument that is portable, rugged, accurate, and may be used easily by regular maintenance personnel. The operator merely places the tip of the welding electrode through the center of the toroid, closes the electrodes, and reads the current flow in amperes directly on the meter.

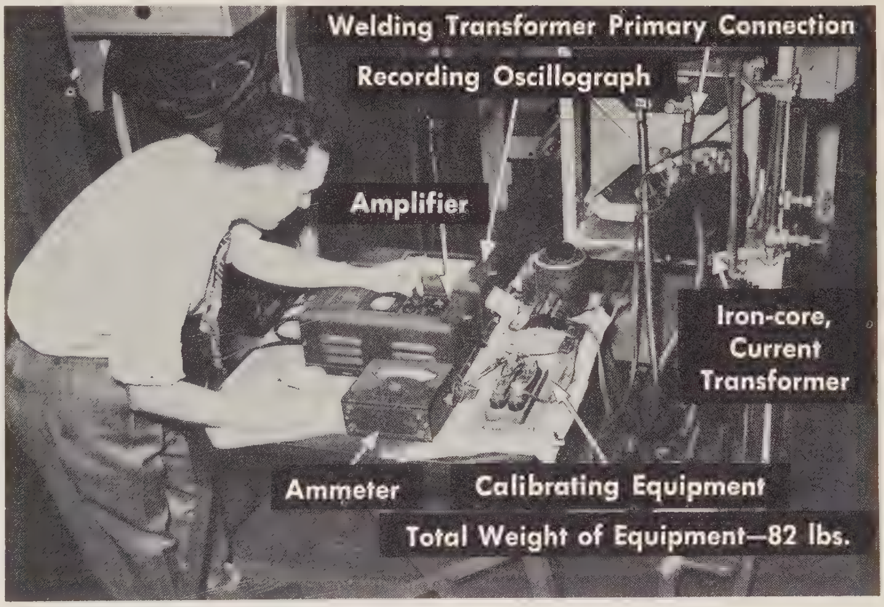

Welding Transformer Primary Connection
Recording Oscillograph
Amplifier
Iron-core, Current Transformer
Ammeter
Calibrating Equipment
Total Weight of Equipment—82 lbs.

Fig. 3—Before the development of the toroidal-type current meter, secondary welding current was determined by measuring the primary current flow with this complex instrumentation and then evaluating the secondary current by approximation. This method was time-consuming and required the services of trained laboratory technicians.

The selector switch has four positions:

- No signal input
- Low range (0 amp to 15,000 amp)
- High range (0 amp to 30,000 amp)
- Calibration.

The first position, no signal input, is a stand-by position to keep the instrument operating but not indicating welding current. A 500-ohm load resistor is connected across the toroid, and the grid condenser is connected to the cathode through a 300-ohm double-tapped, adjustable range resistor. In the low range position, the load resistance is composed of a 1,000-ohm adjustable resistor in series with the tapped 300-ohm adjustable resistor, and the grid condenser is connected to the upper tap of the 300-ohm resistor. In the high range position, the load resistance consists of another 1,000-ohm adjustable resistor in series with the tapped 300-ohm resistor, but the grid condenser is tapped off at the lower tap of the 300-ohm resistor. The variable resistors permit adjustment between the low and the high ranges so that the high range is exactly twice the low range. The calibration position connects the grid condenser to the range resistor and permits the application of an a-c calibrating voltage to the grid condenser to simulate the toroidal output voltage. The zero-reset circuit connects a 10-megohm resistor across the grid condenser to discharge it after each reading.

The 10,000-ohm fixed resistor and 70,000-ohm potentiometer in series with the milliammeter in the plate circuit of the vacuum tube serve to limit and adjust the plate current to precisely 1.0 ma. The 70,000-ohm potentiometer is, therefore, the current zero adjustment. The power supply for the plate and screen grid voltages consists of:

- An isolating transformer with a one-to-one turn ratio
- A bridge-type, full-wave rectifier circuit using four selenium rectifiers
- A pi-type filter circuit employing a 300-ohm and a 150-ohm resistor and two 40-microfarad condensers
- An 0B2, cold-cathode, mercury-vapor, regulating tube.

This circuit has sufficient regulation to permit a 15-v deviation from any normal 115-v, a-c source without affecting the plate current of the vacuum tube.

The circuit to supply the filament voltage to the tube consists of:

- The 6.3-v, a-c, center-tapped winding on the power transformer
- Two germanium crystal diodes in a full-wave rectifier circuit
- A 30-ohm adjustable resistor
- A 50-microfarad filter condenser.

The output of this circuit is from 0.85 v to 0.90 v d-c. The filament voltage of the 1L4 tube is critical since the filament acts as the cathode as well. Operating the filament at lower than the rated 1.4 v gives better tube characteristics and permits measuring lower welding currents, since the cathode-to-ground potential is the limiting factor on the minimum toroidal voltage which will effect a change in the grid bias. The adjustment of filament voltage is provided to compensate for differences in tube characteristics; too low a filament voltage will result in flicker of the meter needle, while too high a filament voltage will result in non-linear tube characteristics.

The calibration circuit utilizes 3.15 v, or half the voltage from the filament winding of the transformer, across the range resistors when the switch is in the calibration position, so that the grid condenser is charged to 3.15 v. This produces a grid bias which results in a certain plate current. A narrow green band is marked on the meter dial and, in the calibrating position of the switch, the meter needle must fall within this band when the instrument is operating correctly. A band, rather than a line indicator, is necessary as the 3.15 calibrating voltage is not regulated and may vary slightly with line voltage variations.

The toroidal pick-up coil used with the instrument was made commercially. It consists of a uniformly wound toroidal coil with 12,000 turns, and has a resistance of approximately 270 ohms, and an inductance of approximately sixty millihenries. The coil is encased in epoxy resin and the connections to it are through a widely used, commercially made, screw-type, coaxial, microphone connector. The cable connecting the toroid to the measuring unit is a low-impedance, parallel-conductor, shielded line to minimize inductive effects, and terminates at the instrument end in a standard phone plug.

Although the welding current meter is actually a peak-current measuring device, the dial is calibrated in root mean square current values, since these indicate the heating effect of the current and are commonly used for welding current readings.

Summary

The development of the welding current meter has resulted in an instrument for the direct measurement of secondary welding currents that satisfactorily fulfills the requirements of accuracy, versatility, portability, and simplicity of operation (Fig. 2). Just how well the instrument satisfies these desirable features is shown by the following characteristics:

- Accuracy—Accuracy is within 3 per cent of the full-scale meter deflection for weld times of from 3 cycles to 30 cycles (60-cycle supply). A weld-time correction curve is provided which serves to increase this accuracy to 1 per cent, if required.
- Versatility—The meter can be used with ½, ⅝, or ¾ in. diameter electrode tips where there is three-fourths of an inch clearance between adjacent electrodes or other parts. The weld time range of from 3 cycles to 30 cycles is adequate for all normal resistance welding operations, as is the current range of from 0 amp to 30,000 amp. Welding current transients (relatively high impulses of current which exist for very short times, usually less than 2 cycles), such as are caused by the initiation or interruption of the welding current—particularly by magnetic contactors arcing severely when the current is broken—do not affect the readings of this instrument.
- Portability—The meter is completely self-contained, measuring only 8 in. by 10½ in. by 11 in. and weighing approximately fourteen pounds. It operates on standard 110-v, 60-cycle, a-c power. By contrast, the equipment previously used as shown in Fig. 3, weighs approximately eighty-two pounds and occupies considerably more space.
- Simplicity of Operation—The meter can be operated by non-technical personnel, since it is a direct-reading instrument; the toroid is placed over an electrode tip and, when a weld is made, the welding current is indicated by the meter. To further simplify accurate readings, the meter circuit has a memory feature; the needle remains at the correct reading for several minutes before gradually shifting toward the zero position. A push button returns the meter to zero between readings. A calibration position of the range switch shows instantly if the instrument is operating correctly.

In addition, the welding current meter is fairly rugged and uses only standard components, except for the toroid. The position of the toroid, when taking readings, is not critical; it may be tipped in any position without affecting the results and it is insensitive to usual surrounding magnetic fields.

Structural Design Problems in Gas Turbine Engines

By PHILIP N. BRIGHT
Allison
Division

Gas turbine engine design for aircraft poses complex engineering problems

The structural design of gas turbine engines for aircraft use is a relatively new field which challenges the engineering prowess of some of the country's finest technical minds. The high power-to-weight ratio, high operating speeds, and searing temperatures common to all aircraft-type gas turbine engines impose design problems which are extremely complex. Additional engineering data, improved metals, and more experience will contribute importantly toward establishing new design criteria which could greatly simplify many perplexing problems in this fascinating field. Already, automatic data-processing equipment is being used to replace the tedious trial and error methods formerly used in compressor and turbine wheel design. High temperature studies have provided a wealth of data which have been helpful in minimizing warpage in new designs of critical components. The continued application of engineering fundamentals and creative imagination ultimately will resolve the present-day intricacies of aircraft gas turbine engine design into relatively straightforward problems in engineering mechanics and strength of materials.

THE structural design of gas turbine engine components has been handled to a large extent by very conservative stress analysis methods. There are several reasons for this—principally the lack of long experience in designing aircraft gas turbine engines, the difficulty in determining the maximum loads to be encountered, and the necessary complexity of the parts which makes stress analysis very difficult.

Structural designers of aircraft components must work to closer limits than most other design engineers. Reliability is the number one objective; then minimum weight, ease of manufacture, and cost follow as design objectives approximately in that order of importance.

Structural Analysis of Gas Turbine Power Plants

Structural analysis involves the determination of the strength and rigidity of a certain structure under specified loading conditions. From the aircraft designer's point of view the real problem is to determine the lightest practical arrangement of material which will transmit the required loads through the specified distance without excessive deflection. Consideration also must be given to the effect of possible vibrations and temperature changes on the structure. With many new high temperature materials presently being developed, the selection of the proper material is sometimes difficult, since physical properties of these new materials are tentative and subject to continual change as more test data become available.

Turbo-prop and turbo-jet engines have similar design problems, except for the turbo-prop reduction gearbox and propeller shafts and the resulting interconnecting structure required to attach the gearbox rigidly to the power sections. The power sections of the turbo-prop engine are practically the same as a turbo-jet engine, except for the additional turbine stages required to absorb the power to drive the propellers. Therefore, the problems discussed apply to both types of gas turbine engines.

There are many facets to the design of a complete aircraft gas turbine power plant. This discussion, however, is limited to some of the structural design aspects of the power section. Such factors as coned discs, external casing stresses, critical speeds and vibrations, and the structural members which interconnect the various components of the gas turbine power plant—while important—are not considered.

Disc Wheel Stresses

Disc stresses for compressor and turbine wheels were determined for Allison Division's first gas turbine engines by using Donath's "Sum and Difference" graphical method[1] and refinements as shown by Stodola[2] which were developed for designing steam turbine wheels. This is a very laborious and tedious method of trial and error, and inclusion of a temperature gradient in the wheel from hub to rim greatly complicates the already difficult problem.

In 1947 the National Advisory Committee for Aeronautics published a paper entitled, "The Determination of Elastic Stresses in Gas Turbine Discs" by S. S. Manson,[3] which explains essentially a finite-difference solution of the equilibrium and compatibility equations for elastic stresses in a symmetrical disc. This method could be adapted with advantage to the International Business Machine (I.B.M.) card programmed calculator, thus eliminating lengthy hand calculations.

In programming the equations for the calculator,[4] certain basic information on the wheel is required. The wheel cross section is first divided into 15 to 20 radial stations and the following values are calculated:

Rim radial stress σ_r due to centrifugal force of blades and rim dovetails:

$$\sigma_r = \frac{mr\omega^2}{A_r} \ (psi)$$

where

m = mass of blade, base, and rim dovetail (lb-sec²/in.)
r = radius to centroid of mass (in.)
ω = angular velocity (rad/sec)
A_r = rim area at base of dovetail or fir-tree slot $= \pi Dh$ (in.²)
Thickness = h (in.)
Mass density × angular velocity squared = $\rho\omega^2$
Poisson's ratio = μ
Elastic modulus = E (psi)
Coefficient of thermal expansion = α (in./in./°F)
Temperature gradient = ΔT (°F).

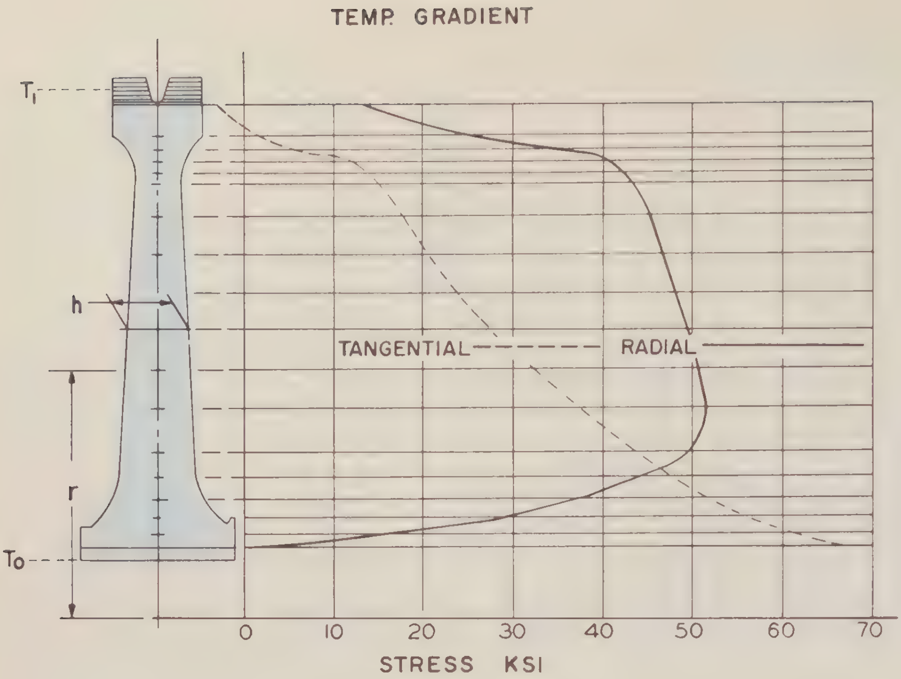
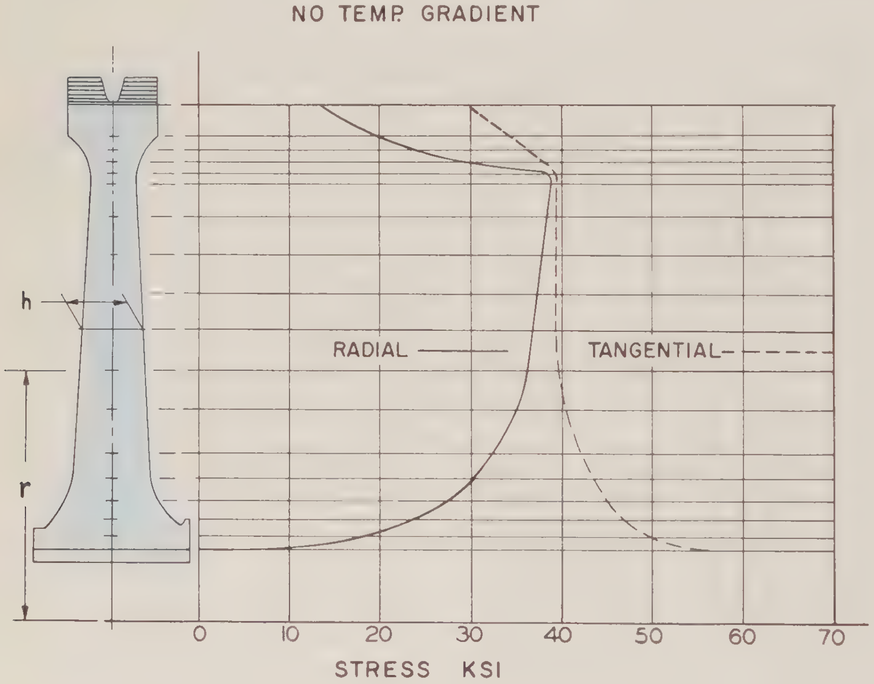


Fig. 1—These graphs show typical turbine wheel radial and tangential stress curves calculated both with and without a temperature gradient between the hub and the outer rim. Automatic data-processing equipment is used to perform the necessary computations, replacing the old, time-consuming trial and error method.



This information is key punched on cards for use in the I.B.M. machine and then the equations are calculated by the card-programmed calculator, producing the following data at each radial station:

Tangential stress = σ_t (psi)
Radial stress = σ_r (psi)
Radial deflection = Δr (in.)
Cross sectional area = A (in.^{2})
Volume = V (in.^{3})
Polar moment of inertia = I_p (lb-ft^2)
Average tangential stress = σT_{avg} (psi).

The tangential and radial stresses a then plotted as shown in Fig. 1. Su factors as slight changes in the wh cross section, changes in rim radial str due to blade changes, and speed chan can be rapidly checked, thus speeding the wheel design.

Further refinement of wheel desi methods is now in prospect which w take the calculations into the plas range, thus enabling such factors as str and deflection to be determined af portions of the wheel have exceeded t material's elastic limit. This method shown in a N.A.C.A. report entitl "Determination of Stresses in Gas Tu bine Discs subject to Plastic Flow a Creep," by M. B. Millenson and S. Manson.[5] This method of wheel desi has not been used by Allison's engine as yet; however, the possibility of furth reducing rotating disc weights makes attractive and the method is bei programmed.

It has been claimed that a wheel w not burst until the entire cross secti reaches the rupture strength of the ma rial. Taking into consideration some the failures encountered this theory open to question, so an attempt was ma by Allison's engineers to determine t conditions when a complete burst occu Figs. 2 and 3 show wheel bursting con tions. The pictures were obtained setting electrical contacts close to t wheel rim so that as the rim expand at the first phase of failure, a high-spe flash photograph was taken.[6]

Blade Retention

There are two basic methods commo used for blade retention—the 45° do tail for compressor blades and the multi serration or "fir tree" for turbine blac (Fig. 4).

Analysis of the blade base and whe rim dovetail slot is by convention methods. Centrifugal forces of the bla and base are assumed equally distribut to each side of the base. For the turbi blades this load is further distributed proportion to the contact length of ea fir-tree lug. This may not be exactly tr but it is believed that approximat equal distribution will result after slig yielding of the most heavily loaded lu As shown in Fig. 4, the critical secti across each lug is analyzed for she loads, making sure to add the centrifu load caused by wheel rim lug support itself, in checking section B-B.

The lug is analyzed for the loads shown in Fig. 4 as follows:

P = centrifugal load of blade and base

P_L = centrifugal load of wheel lug

P_N = normal force between blade base and lug = $P/2 \sin\theta$

P_C = compressive force (tangential) in wheel lugs = $P/2 \tan\theta$

θ = angle of blade base contacting wheel lug

a = arm from centroid of applied load to critical lug section

b = width of minimum lug tension section

A = area of cross section at section being analyzed.

Section C-C is considered critical principally in shear with $f_s = P/2A_c$ (area at section C-C). Section A-A is analyzed for combined shear, bending, and compression using the following expressions:

$$f_{s_{max}} = \left[\left(\frac{f_b}{2}\right)^2 + (f_s)^2\right]^{1/2}$$

$$f_{n_{max}} = f_{s_{max}} + f_b/2$$

where

f_s = shear stress = $P/2A$

f_b = bending stress + compressive stress

$$= \frac{MC}{I} + P_c/A$$

$f_{s_{max}}$ = maximum combined shear stress

$f_{n_{max}}$ = maximum combined normal stress.

This is considered conservative since the beam is very short for its depth and, therefore, the conventional flexural beam theory MC/I is not strictly valid. A stress concentration factor is added, depending on dimensions of the groove.[7] Allowable stresses used are for the maximum temperature expected and a margin of safety of approximately twenty-five per cent (based on ultimate) is maintained in order to allow for vibratory stresses.

Photoelasticity might well be employed to investigate this problem further in order to determine such factors as stress concentrations and stress distribution and make possible further refinements. Reduc-

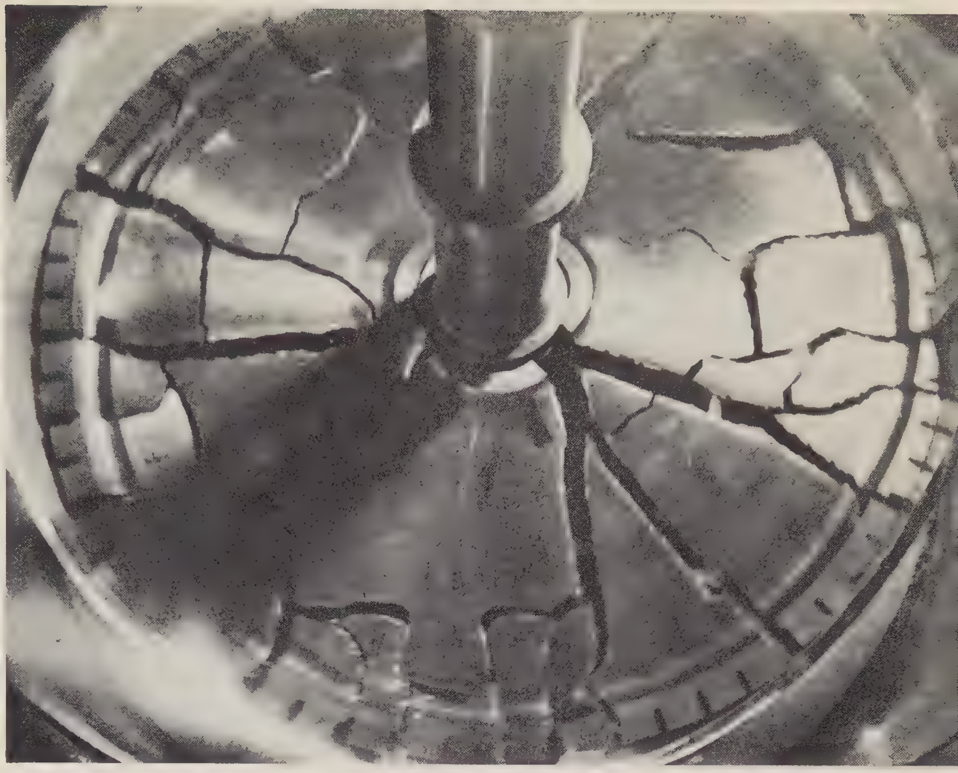


Fig. 2—This wheel, shown bursting in a spin pit (burst speed is 21,800 rpm; rim velocity is 1,720 ft/sec), is composed of titanium alloy. Early wheel-burst tests made by Allison Division's engineers were conducted using several different alloys of titanium for the purpose of checking relative bursting speeds. Note that the failures appear to be instantaneous in both radial and tangential directions, indicating possibly equal rupture stresses (biaxially) over the disc.

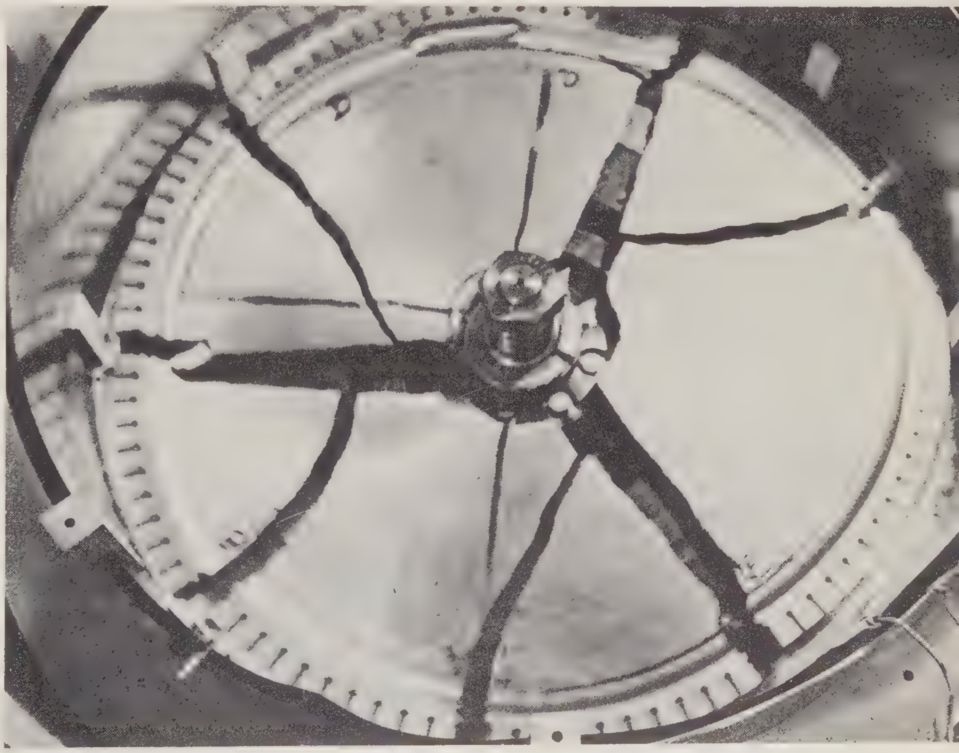


Fig. 3—A low-carbon, brazed compressor wheel, when subjected to a burst test (burst speed is 15,100 rpm; rim velocity is 1,040 ft/sec), shows that failures are predominately radial, indicating excessive tangential stresses. However, the failures are all very brittle in appearance, indicating also that the biaxial stress level approaches the rupture strength of the material fairly equally over the disc after the yield stress is exceeded.

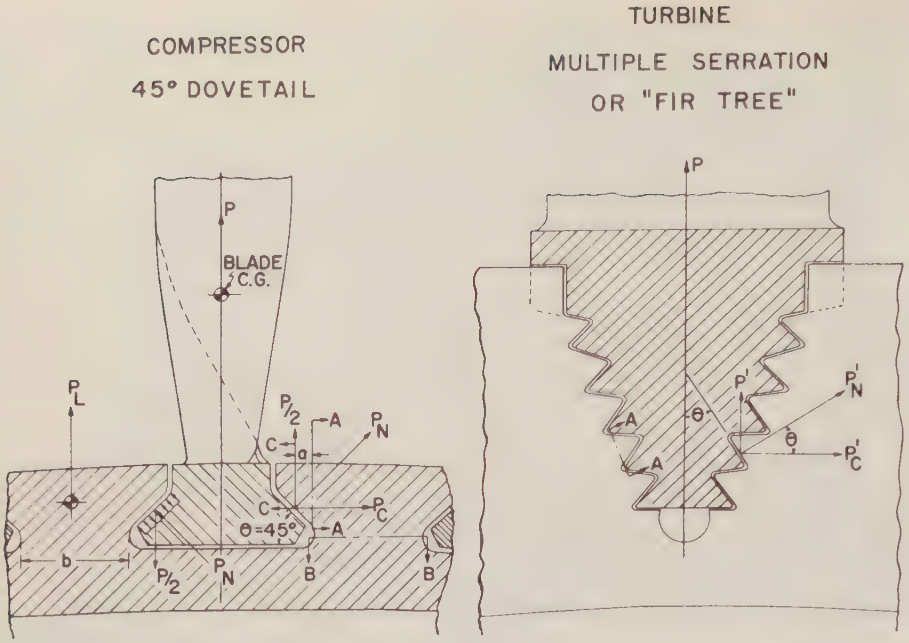


Fig. 4—The original practice regarding blade retention was to machine and fit the blades very closely with a tight selective fit, but present practice is to allow more tolerance and clearance in machining the dovetails and blade bases so that the blade is loose when installed. This reduces extremely close machining of base and wheel-rim slots and allows the blade to seek its own radial position (within limits) as centrifugal force increases. The former bending stresses on the blade base from blade tilt caused by machining tolerance and a tight fit are thereby relieved, and the major stress is produced by the centrifugal force of the blade. Bending stresses from gas loads on the blade also are reduced due to the centrifugal force so that the gas load produces only slight bending and a shear stress at the blade root which is negligible compared to the centrifugal load.

tion of the number of fir-tree lugs on turbine blades would be a definite advantage and work along this line is in progress.

Bearing Supports

The problem of transferring loads from the rotor bearings to the outer shell has been one of the most troublesome. A spoked wheel structure of some kind is necessary since the loads must pass through the annular hot air or gas passage without causing appreciable restriction to the flow. To cut down on flow restriction, the spokes are streamlined. Difficulties have occurred principally in the turbine section of the engines where a large temperature gradient exists between the inner hub and outer shell (Fig. 5).

Stress analysis indicated that only the static stresses of large magnitude were from thermal expansion. The temperature of the outer shell, being 250°F to 300° F higher than the inner bearing housing, caused the outer shell to attempt to enlarge in diameter faster than the spokes and inner housing would allow. This resulted in high bending stresses on each side of the weld tying the spokes to the outer shell and a corresponding tensile load in the spokes. Bending stresses in the inner bearing housing were not critical because of the heavier section and smaller diameter.

By determining the free expansion of the outer shell, the struts, and the inner bearing housing due to temperature change and writing equations for the deflection of each under the common load W (in each strut) and setting the sum of these deflections equal to the differential free expansion, the load W can be determined[7] in the following manner:

$$\mathcal{E}\text{ struts} = \Delta T \times l \times \alpha$$
$$\mathcal{E}\text{ housing} = \Delta T \times R_2 \times \alpha$$
$$\mathcal{E}\text{ outer shell} = \Delta T \times R_1 \times \alpha$$
$$\mathcal{E} = \mathcal{E}\text{ outer shell} - \mathcal{E}\text{ housing} - \mathcal{E}\text{ struts}$$

$$W = \frac{\mathcal{E}E}{f(\theta)\left[\frac{(R_1)^3}{2I_1} + \frac{(R_2)^3}{2I_2}\right] + \frac{l}{A}}$$

where

W = load in struts (lb)

$\mathcal{E}$ = total differential deflection—per radius—from thermal expansion (in.)

E = modulus of elasticity—all materials similar (psi)

$f(\theta)$ = term depending on spacing of struts

$$= \frac{1}{\sin^2\theta}\left[\frac{\theta}{2} + \frac{\sin\theta\cos\theta}{2}\right] - \frac{1}{\theta}$$

Note: 2θ = angle between struts

ΔT = change in temperature of part (°F)

R_1 = radius to centroid of outer shell or rim (in.)

R_2 = radius to centroid of inner bearing housing (in.)

I_1 = moment of inertia of outer shell cross section (in.^{4})

I_2 = moment of inertia of inner bearing housing cross section (in.^{4})

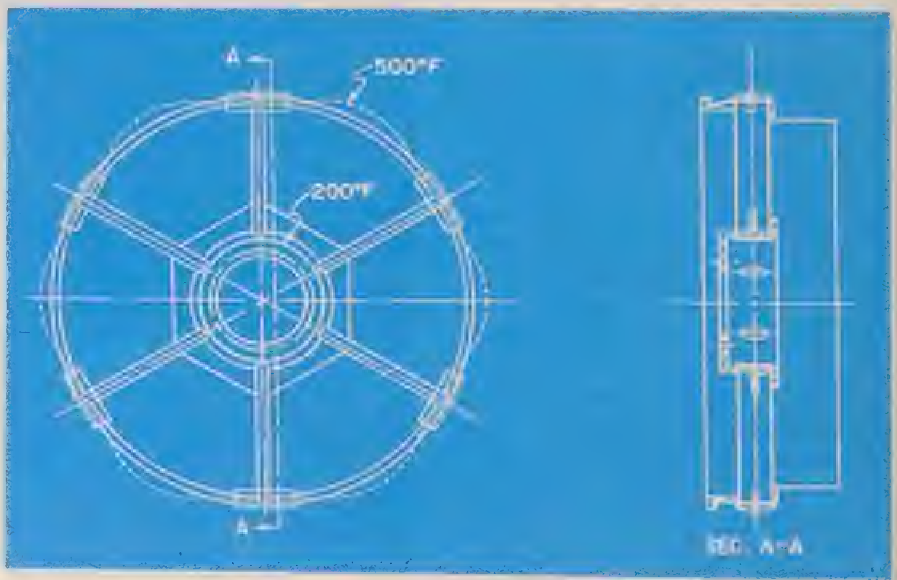


Fig. 5—The first turbine bearing support assemblies were rigid welded assemblies with gussets welding the struts or spokes to the inner bearing housing and outer shell. A combination of thermal expansion and vibration caused cracking at the ends of the struts or gussets. Enlarging the gussets merely moved the stress concentration point to the edge of the larger gusset and cracks again resulted.

l = effective length of struts (in.)
A = cross sectional area of strut (in.^{2})
α = coefficient of thermal expansion (in./in./°F).

The maximum bending moments and stresses may be determined in each member by the following expressions:

$$\max +M = \frac{WR}{2}\left[\frac{1}{s} - \frac{1}{\theta}\right] \text{ between struts}$$

$$\max -M = -\frac{WR}{2}(1/\theta - \cot\theta) \text{ at struts}$$

maximum tension or compression

$$= \pm\, 1/2\, W\left[\frac{1}{s}\right] \text{ (circumferential in outer shell)}$$

where

$R = R_1$ or R_2 above
$s = \sin\theta$.

The critical point, of course, is in the outer shell at the attachment of the strut where bending stresses build up rapidly for moderate loads in the strut.

The first attempt to reduce these stresses was to reduce the stiffness of the outer shell or rim by scalloping the flange between struts, thereby reducing the load in the strut and lowering the bending stresses in the outer shell. This was abandoned quickly when vibration difficulties increased due to reduced stiffness.

Another design attempted to eliminate the strut tension loads caused by thermal expansion by sloping the struts so that they attached to the bearing housing tangentially (Fig. 6).

In this type of bearing support, as well as the radial type, the direct bearing loads are distributed to the spokes in proportion to the relative stiffness of each spoke by taking into consideration the following:

- Spoke stiffness in tension or compression and bending
- Bearing housing stiffness from unsymmetrical strut loads
- Outer shell stiffness from unsymmetrical strut loads.

By applying a unit load in the direction of the applied load to each spoke and writing expressions for the deflections of the spoke, housing, and outer shell, the load distribution between the spokes can be determined. Then, by superimposing these loads on those loads due to thermal expansion, the resultant loads are determined throughout the assembly and the stresses calculated.

Trouble was encountered in holding dimensions and preventing warpage during welding and heat treating operations. These fabrication difficulties have limited the use of this type of bearing support but further development should produce a satisfactory structure.

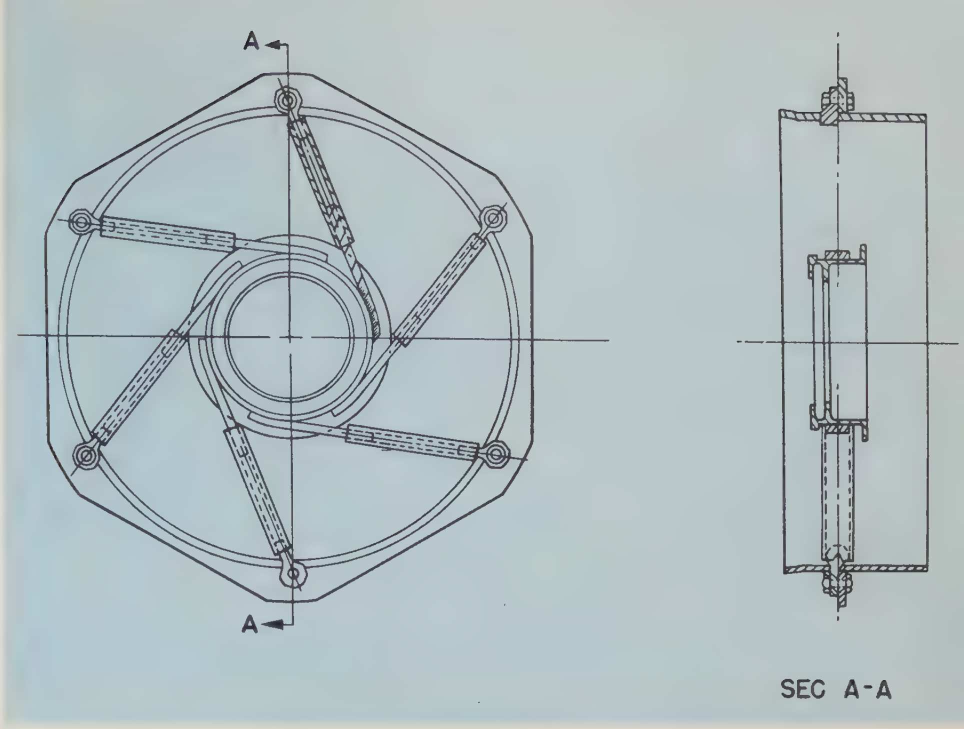


Fig. 6—In an attempt to eliminate strut tension loads caused by thermal expansion, the struts were sloped so that they attached to the bearing housing tangentially. With this arrangement the differential expansion causes a slight rotation of the bearing housing with respect to the outer shell, and flexible joints may be used to reduce the resultant bending moments at the ends of the struts. Direct bearing loads from gyroscopic and g loadings are carried by the struts approximately in proportion to their slopes.

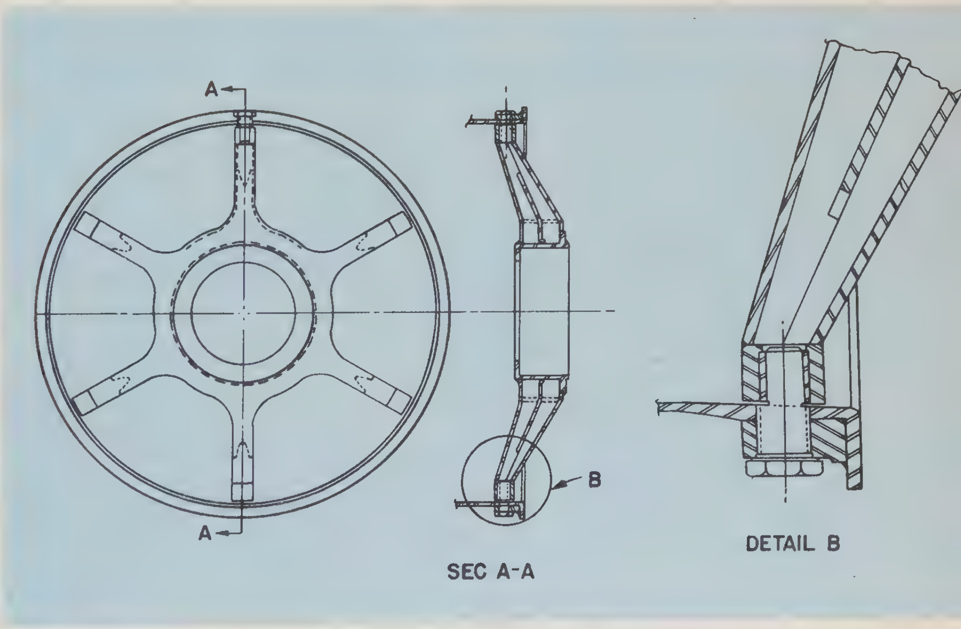


Fig. 7—Another form of turbine bearing support makes use of sliding pins at the ends of the supporting struts to allow for differential thermal growth.

A third design using a different principle appears to hold some promise (Fig. 7). The bearing housing and spokes are one welded assembly with sockets in the ends of the spokes. The outer shell is a separate welded assembly with bosses containing removable pins. These pins slide in the sockets in the spokes when differential expansion occurs. By holding very close tolerances the sliding deflection for direct loads can be held to a minimum and stresses from thermal expansion are thereby eliminated. In transmitting bearing loads from gyroscopic

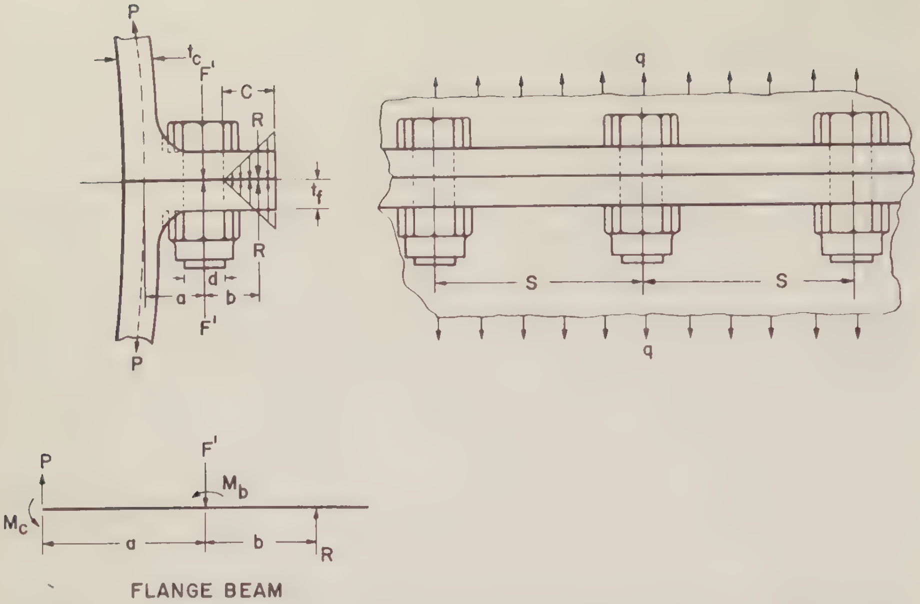


Fig. 8—A typical longitudinal split-line flange on a compressor casing showing loading from internal pressure. The basic dimensions shown are required for stress analysis.

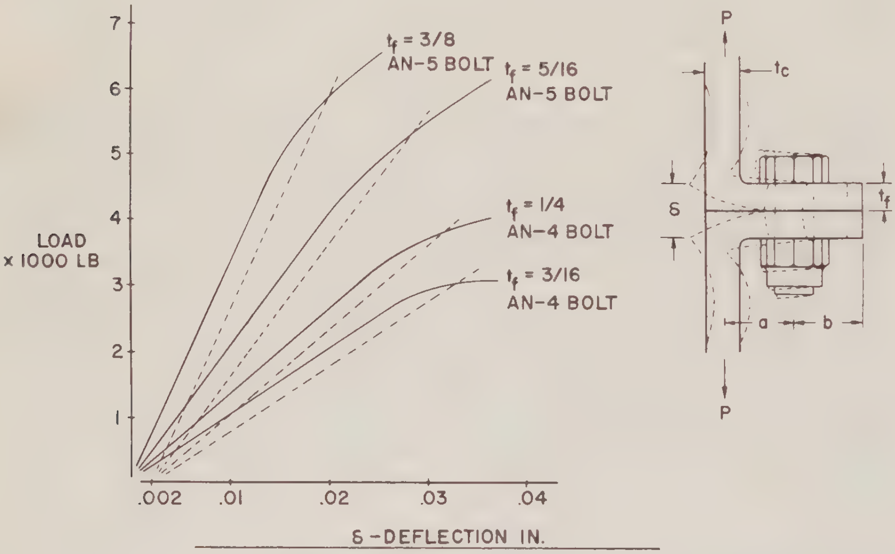


Fig. 9—In order to check the stress analysis technique used in designing longitudinal split-line flanges, static tests were run in a test machine on a series of samples of bolted flanges wide enough to contain four attaching bolts. The curves shown are plots of load versus deflection at the heel of the flange. The material used in the test specimens was annealed-type 347 stainless steel and, therefore, some of the load-deflection curves do not have a uniform slope in the elastic portion of the curve, except where thick flanges caused high tensile stresses in the bolts. It was necessary to estimate the "effective" slope in order to determine the yield strength of the test piece. Calculated yield strength using the method described in the text is shown by the test results to be reasonably accurate, considering the type of material being tested.

and g loadings, the spokes in the direction of loading carry no load and, therefore, the side spokes must transfer the load in bending to the outer shell. This requires a spoke which is stiffer in bending in a lateral direction, since all loads between the pins and sockets will be normal to the spokes.

The principal difficulty with the first design of this type was the rapid wear of the pins in the sockets, resulting in loss of the necessary close tolerance. A redesign with larger, extremely hard, and possibly lubricated pins should lower the bearing stresses to a point that may be satisfactory.

Split-Line Flange Stresses

Split-line flanges can be designed from fairly straightforward stress analysis methods. While admittedly not exact, the following method gave results that were substantiated reasonably well with static tests.

The more critical type of split-line flange is the longitudinal, since it can readily be seen that the radial type has an inherent advantage due to curvature. In addition, hoop tension loads from internal pressure must be carried across the longitudinal split-lines and only a smaller, internal pressure load across the radials. Therefore, the following method is for a longitudinal split-line typical of axial flow engines.

Fig. 8 shows a typical longitudinal split-line on a compressor casing where

q = tension load in case—hoop tension (lb per in.)
S = bolt spacing (in.)
P = load per bolt = qS (lb)
t_c = case thickness (in.)
t_f = flange thickness (in.)
a = case centroid to bolt ℄ (in.)
b = bolt ℄ to flange reaction R (in.)
c = estimated flange compressive width (in.)
d = bolt diameter (in.)
d' = root thread diameter (in.)
F = preliminary resultant bolt load (lb)
F' = resultant bolt load (lb)
R = flange reaction—resultant force at centroid of triangular load distribution (lb)
p = internal pressure (psi)
r = radius of case (in.).

The following calculations are used for a preliminary determination of bolt size:

- Hoop tension load in case from internal pressure = $q = pr$

$$P = qS$$

$$F = \frac{P \times (a+b)}{b}$$

- Bolt size required with an ultimate strength = $3F$
- Initial torque on nut to stress bolt to one-third ultimate tensile strength.

To calculate the moment distribution between the flange and case, it is necessary to take into consideration that the flange and bolts must act together in bending and it can be conservatively assumed that the maximum moment occurs at the bolt. The total moment Pa is distributed between the flange and case in proportion to their moments of inertia. Therefore,

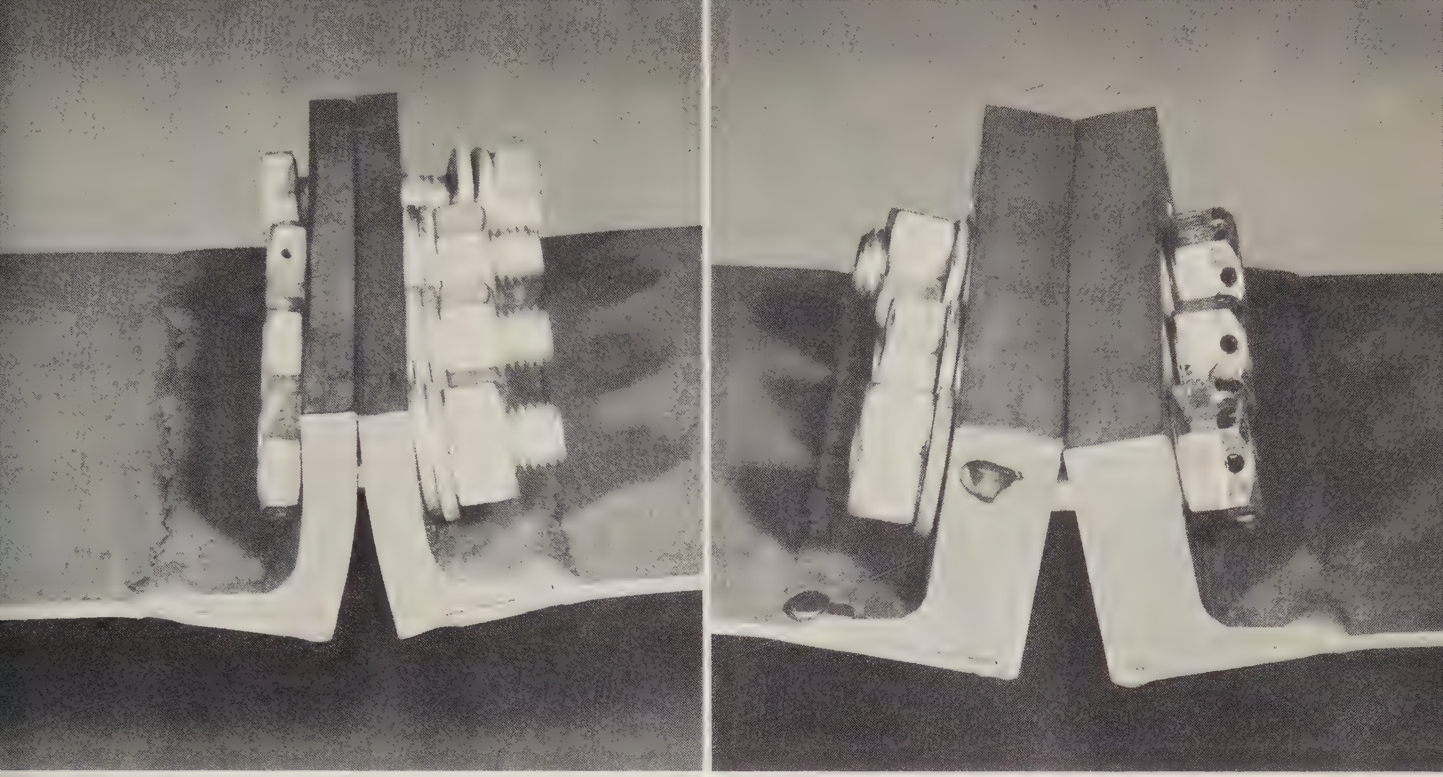


Fig. 10—Typical four-bolt test specimens representing a combustion section longitudinal split-line flange.

$M = Pa =$ total moment (in.-lb)

$I_{fb} =$ moment of inertia of flange and bolt $= I_f + I_b$

where

$I_f = St_f^3/12 - dt_f^3/12$ and $I_b = 0.0491\,(d')^4$

$I_c =$ moment of inertia of case $St_c^3/12$

$M_{fb} =$ moment carried by flange and bolt $= Pa\,(I_{fb}/I_{fb} + I_c)$

$M_c =$ moment carried by case $= Pa - M_{fb}$.

To calculate the distribution of M_{fb} between the flange and bolt,

$M_f = M_{fb}\,(I_f/I_{fb})$

$M_b = M_{fb} - M_f$.

Referring to the flange beam in Fig. 8,

$M_R = P(a+b) - F'b - M_c - M_b = 0$

$F' = P(a+b) - M_c - M_b/b$.

To calculate the bolt stresses,

$f_{bolt} = f_t + f_b$

$f_b = M_b(d')/2I_b$

$f_t = 4F'/\pi(d')^2$.

(Note: Increase bolt size or heat treatment calculated in the preliminary determination of bolt size if $f_{bolt} \times 1.15$ exceeds yield strength F_{ty} of bolt material.)

To calculate the flange stresses,

$f_b = M_f(t_f)/2I_f$

$$f_s = \frac{P}{S(t_f)}$$

$$f_{s_{max}} = \left[f_s^2 + (f_b/2)^2 \right]^{1/2}$$

$$f_{n_{max}} = f_{s_{max}} + f_b/2.$$

To calculate the case stresses

$f_{case} = f_b + f_t$

$f_b = M_c(t_c)/2I_c$

$f_t = P/S(t_c)$.

(Note: This method assumes rectangular sections for flange and case. For other shapes the section properties of the flange and case must be calculated and the proper values used in the stress formulae.)

This method agrees reasonably well with test results (Figs. 9 and 10) when:

- Flange thickness is approximately equal to bolt diameter
- Ratio of *a* to *b* is approximately 1
- Bolt spacing is approximately 4*d*.

A new design of this type of flange is needed. However, difficult fabrication processes have ruled out many of the proposals so far.

Summary

It is believed that the future of the gas turbine engine for aircraft depends upon the imagination and ingenuity of today's and tomorrow's engineers. While the structural design problems this type of power plant poses today are highly complex, the sound application of engineering fundamentals in developing new methods coupled with new developments in the field of metallurgy will resolve these problems and will result in a steady increase in the already excellent power-to-weight ratio.

Bibliography

1. Donath, *Die Berechnung Rotierenden Scheiben und Ringe* (Julius Springer, Berlin, 1912).
2. Stodola, Dr. A., *Steam and Gas Turbines* (New York: McGraw-Hill Book Company, Inc., 1927; reprinted Peter Smith, New York, 1945).
3. Manson, S. S., "The Determination of Elastic Stresses in Gas Turbine Discs," *National Advisory Committee for Aeronautics—Technical Notes*, No. 1279, 1947.
4. Rea, J. C., "Instruction Manual for Preparation of Problems for Running on the Card Programmed Calculator," Engineering Report No. 473, Allison Division, General Motors Corporation.
5. Millenson, M. B. and Manson, S. S., "Determination of Stresses in Gas Turbine Discs Subject to Plastic Flow and Creep," *N.A.C.A.—Technical Notes*, No. 1636, 1948.
6. Allison High Speed Flash Photography Equipment, General Motors Research Laboratories Division, Report No. 32-380.
7. Roark, R. J., *Formulas for Stress and Strain* (New York: McGraw-Hill Book Company, Inc., 2nd Edition, 1943).

How to Organize and Write Effective Technical Reports

It is important that engineers know how to write effective technical reports. Engineering data not adequately reported have no permanent value except to the person who originally did the work. Thus, one of the most important tasks of the engineer is to produce technical writings that will be read, understood, and used by both technical and non-technical people. Most engineers can become good writers if they will put the same thought and effort into this phase of their work as they do in conducting their experiments.

The very nature of the engineering profession dictates that those who practice it must be able to report their findings in a clear, concise, and accurate manner in order to insure the best utilization of the data being reported and to contribute significantly to the ever-expanding store of technical knowledge. There is a great deal of evidence that engineers do write and, as a matter of fact, with a considerable degree of proficiency. This is evidenced by the ever-increasing size of The Engineering Index and the many fine articles which appear in engineering periodicals. Moreover, the quality of the writings in the technical literature during the past 10 or 15 years has steadily improved. The engineering colleges and universities are to be commended for the outstanding work they have done over the past decade to effect this improvement.

Engineers too often think their part of the job is finished when they complete their tests, drawings, or designs. The part of the work that makes the job useful or successful is frequently an analysis of the results presented in a well-prepared report. One of the very important jobs of the engineer is, therefore, to produce technical writings that will be read, understood, and used by all groups, even though they are probably not familiar with the subject. This means that there must be a large element of sales and educational effort in any technical writing. Hours spent in writing may well prove to be an engineer's most productive time. Moreover, from a selfish standpoint, the engineer or scientist who can do an effective job of communicating his ideas to others—particularly to those in management—usually advances up the professional ladder somewhat faster than his colleagues who have not acquired this ability.

What constitutes a good technical report? The following general comments may help to answer that question:

- The text should be neither ambiguous nor subject to misinterpretation by any reader, regardless of his technical background.
- The report should reflect sound, objective thinking.
- The content should be factual and easily read. Word choice should lean toward simple rather than complex words.
- A report demands and is strengthened by the use of the correct word. Knowledge of the language and its principles of usage will aid in the expression of the writer's thoughts.
- The neat and orderly report commands respect.

Kinds of Technical Reports

Generally speaking, technical report writing falls into four categories: (a) letters, (b) memorandum reports, (c) progress reports, and (d) final reports.

Letters

It may seem strange that letters are considered a form of technical reporting. Often, however, they include important data or facts, and it is not uncommon to see a project engineer thumbing through his correspondence file for letters written to associates concerning a particular project or investigation which may help him in the preparation of a progress report or final report.

Memorandum Reports

The memorandum report is usually written on a standard form for limited distribution within the organization. It is purposely kept as short and to the point as is practical to get pertinent information about a particular project o
paper as quickly as possible and di
tributed to people in the organizatio
who can use it effectively in connectio
with their own work.

Progress Reports

The progress report generally is muc
longer and more formal in its style an
format than the memorandum. Usuall
a progress report is made when work o
a project has reached the end of a pa
ticular phase or at regular intervals s
by management. For example, a long
range project of possibly two or thre
years' duration should produce a pro
ress report at the end of the prelimina
investigation or literature-searchin
phase. At this point it would be decide
whether or not to continue the project c
the strength of the information u
covered. Progress reports many tim
are prepared to recommend steps i
future planning or in some cases t
indicate results that may be forthcomin
Such reports also keep top manageme
informed regularly of the activities goi
on in the engineering department.

Final Reports

The final report, as its name implie
is the last to be written concerning
particular project or investigation. It
often quite long and covers in minu
detail all the salient facts about t
project. A great deal of care must b
exercised in writing the final report as
becomes a permanent record, and t
conclusions, recommendations, and da
it contains may be used for many yea
as a reference for making decisions a
in gathering material on new investig
tions. The final report should, therefo
be as complete as it is possible to make

Almost all organizations that empl
engineers and other technical people u
a form of report similar to one of the

By **RALPH A. RICHARDSON**
Research Laboratories Division
and **C. A. BROWN**
General Motors Institute

Good technical reporting is the cornerstone of engineering progress

Experience has shown that the engineer who performed the experiments should be responsible for writing the report, since he is the only one who knows all of the details.

Organizing the Technical Report

There are many ways of organizing and writing a technical report. The method used will depend upon the use to which the report will be put and the kind of subject matter it contains. In other words, the kinds of readers and what they need to get from the report will govern the organization and language. In addition, there are sometimes "ground rules" of the organization that must be observed.

No matter who the readers are they will want to know early in the report what it is about, the extent of the coverage, and the most significant information contained in the report. For this reason, reports that are predominantly informational are oftentimes preceded by a summary or abstract to provide the user with a quick understanding of the content. Incidentally, the validity of such a report rests upon its completeness, accuracy, and ease of use. On the other hand, a report that gives the outcome of a study, an investigation, or research must provide a clear statement of the problem, an explanation of the method employed in solving the problem, and a clear statement of the results or conclusions.

One method of organization that is used in reporting a piece of research or information has evolved through years of experience and has proved to be a highly effective way of presenting technical material. The general organization of the material is as follows:

- Title
- Table of contents
- Opening
- Conclusions
- Recommendations
- Body of the report
- Reference list.

Although this sequence differs somewhat from the common practice of technical report writing in many organizations, it is recommended primarily because it is easier for the reader to grasp the problem and conclusions of the investigation without having to read through the entire manuscript, and leaves him free to peruse the details at his leisure. One of the first steps in an engineering investigation is to make a search of the literature in the field of the investigation to find out what others have done on the same or similar problems. It might be mentioned here that all engineers should be familiar with library services and how to use them efficiently. The next step is to select a suitable title for the work.

Title

The title should indicate the contents of the report to the prospective readers by being as descriptive as possible. Some authorities oppose this view, maintaining that very often the title becomes too long and involved; but it would seem that for technical reports, at least, the advantages of a descriptive title outweigh the disadvantages. Recent practice in many organizations is to give the report a short title composed of key words followed by a longer, more descriptive subtitle.

Table of Contents

A table of contents is included in most engineering reports to help the reader locate quickly and easily the main sections and subsections of the report and to give an idea of overall content. Usually, however, in the case of relatively short reports the contents page is omitted.

The table of contents is usually prepared from the author's outline used in writing the report, since it lists in detail all of the important points covered in the text. It might be well at this point to mention that a well constructed outline is essential to good report writing. While it may seem that the preparation of an outline is a non-productive effort, experience has shown it insures that all points are discussed, that the chances of leaving something out are minimized, and that the writing is kept on target. Generally speaking, it is not necessary to outline the opening, conclusions, and recommendations unless these sections are long and complex. As a general rule, however, it is better practice to keep them as short as possible.

Opening

The opening of an engineering report in some cases is called a foreword and in others the introduction. In either case, however, the opening should be a clear-cut statement of the problem or investigation undertaken and should be given careful thought. Many engineers make an entry in their logbook on the day the project is assigned, stating briefly what they are setting out to accomplish. Very often this log entry serves as the opening to their report. There is considerable merit in doing this because engineers, like other people, are human and sometimes have a tendency over a period of time to lose sight of the specific objectives of a particular project. Reference to the logbook and to the brief statement of the problem as expressed on the day it was assigned helps not only to keep the objective in mind, but also proves helpful when starting to write the report.

Conclusions

The conclusions should follow directly after the opening of the report and, in general, should point out what the engineer found out about the problem stated in the opening. Although the conclusions are sometimes listed numerically, they need not be; however, each should be separate and distinct from the other so as not to confuse the reader about the points being emphasized. Each conclusion should be a specific, positive statement and should leave no doubt as to its true meaning.

Recommendations

An author's recommendations very often can be the most important part of a technical report, and their adoption or rejection depends to a large extent on the manner in which they are presented. Like the conclusions, they should be positive statements suggesting some specific thing to be done or course to be followed, and should include an estimate of the cost involved, if any. Few engineers like to be concerned with how much it costs to do things, but it is almost a sure bet that the first question asked after reading the recommendations in a report will be, "How much will this cost?" It is good report writing technique to anticipate this question by giving solid figures based on the best estimates that can be made.

If this is done, the chances of the recommendations being accepted will be greatly enhanced. Moreover, this technique of anticipating questions should be considered not only with regard to costs, but with regard to every phase of the report. It is a practice that is highly recommended.

These three parts, the opening, conclusions, and recommendations are the most difficult for the majority of technical men to write satisfactorily because they must summarize the work in simple language. They are written for the person who does not have enough time to familiarize himself with the many details of the subject—particularly for the executive who must make engineering decisions.

Discussion

The body or discussion of an engineering report is generally the largest single section. In it are included details of the investigation such as:

- Text which supports all conclusions and recommendations
- The kind and accuracy of the instrumentation used
- Test setup and problems encountered in this phase of the work
- Detailed explanation of the test procedure
- Presentation of data taken along with an interpretation and analysis of these data.

Illustrations such as curves, photographs, charts, and tables should be included to enrich the discussion. (Details as to handling illustrative material are discussed under the heading *Illustrations* in this paper.)

Usually, the biggest problem in connection with the body or discussion section of the engineering report has to do with organization of the material being presented. Care should be taken to arrange the material in a logical manner so that a reader unfamiliar with the subject can readily grasp the meaning of what has been written. Thus, the first subheading in a typical report might be *Historical* or *Background Information*. A few opening remarks outlining the project and pointing out some of the thinking leading up to the work are very helpful in orienting the reader to the problems and objectives of the investigation. Following this, the information gathered should be presented as simply and as straightforwardly as possible, avoiding trite expressions and techniques which lower the literary and informational quality of the report.

While this paper is primarily concerned with the organization of material and the mechanics of technical report writing, it should be emphasized that the application of the rules of good grammar and composition is a prerequisite to quality report writing. This does not mean that every engineer is expected to be an expert writer, but does assume that he is well grounded in grammar, spelling, and the rules of good composition.

A generous use of subtitles throughout the discussion helps the reader to visualize more completely the overall picture or scope of the work, as well as to gain a more lasting impression of the material presented. Moreover, subtitles tend to break up long expanses of straight text which very often are dull and difficult to read. Subtitles also tend to dramatize the material and improve greatly the overall quality and readability of a technical report.

Reference List

The reference list is an important part of the technical report and much care should be given to its preparation. This should consist of two lists: (a) the periodical list and (b) the book list. References are best listed alphabetically by author, each entry to include the title of the article or book, the volume and number in the case of periodicals, the publisher in the case of books, the date of publication, and finally page references. Only those works which the author has read and feels are pertinent to the subject of the report should be included in the reference list. Numerous titles of books and periodicals listed merely to impress the reader or to add bulk to the manuscript most often accomplish the opposite effect.

Illustrations

Illustrations are included in an engineering report to enrich the text as well as to make it easier to understand. Like the reference list, illustrative material should not be used just for the sake of having it or to increase the bulk of the report. Nor should it be used as a crutch for poorly written material. Rather, illustrations should be looked upon as aids in explaining points covered in the text. Good illustrations not only enhance the appearance of a report, but they also aid greatly in making more lasting impressions upon the reader than do words alone.

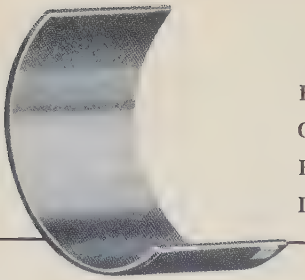
Illustrations include such things as photographs, drawings, curves, sketches, and tables of data. Very often the discussion section of a technical report is devoted almost entirely to the interpretation of illustrative material. Many authors, however, experience difficulty in deciding what and how many illustrations to use. A general rule to follow in this connection is to use illustrations only when they are necessary to support the thoughts, ideas, or points in the report. Many engineering reports are inadequate because of a lack of good quality illustrative material.

Illustrations should be large enough to convey the message clearly and quickly. Too often, size is sacrificed for economy reasons. Authors sometimes select small photographs or print too many illustrations on a page. Pictures should be reproduced large enough so that all pertinent details are readily apparent. Sometimes, photographs are retouched to highlight or accent a particular area which helps to clarify points discussed in the text. Curves which graphically illustrate engineering data should be plotted neatly and reproduced page size, if possible, particularly if the author has prepared them with the idea they will be used for "picking off" values. Sketches and drawings, like photographs, always should be large enough to show clearly all necessary details.

Summary

Success in engineering, as in all other organized undertakings, depends as much on the ability to present an idea convincingly as it does upon the ability to perform experiments and calculations. The most miraculous discovery in the laboratory is not a contribution to the store of knowledge until the results are recorded and transmitted to others. A written report is often the only record which is made of the results that have come out of years of thought and effort. It is used to judge the value of the engineer's work and serves as the foundation for all future action on the project. If the report is written clearly and precisely, it is accepted as the result of sound reasoning and careful observation. If the report is poorly written, the results presented in it are often dismissed as the work of a careless or incompetent worker. Most engineers can become good writers if they will put the same thought and effort into this phase of their work as they do in conducting their experiments.

Why Bearings Seize



By **ARVID E. ROACH** and
CARL L. GOODZEIT
Research Laboratories
Division

New bearing
theory clarifies
empirical data

When metals slide against each other at high speed and under heavy load, they tend to seize or weld together. Engineers are not aware of any pair of metals that is entirely free from this tendency; all metals, in the absence of surface films, can be made to weld together to a greater or lesser degree. Fortunately, there are certain pairs of metals that weld together less readily than others. These are the *good* bearing metals—a class of materials whose importance in industry can scarcely be overestimated. Without them, all machines would grind to a standstill, their contacting parts hopelessly stuck together. Experiments performed with forty different elemental metals at the Research Laboratories Division have confirmed a new bearing metal theory which clarifies and unifies much that has been learned empirically about bearing materials, and provides a useful basis for the selection of new materials for bearing applications. It has been found that a good pair of bearing metals consists of those metals which have atomic sizes that are so different that few junctions are formed between them when they are rubbed together. Not only should this condition be satisfied, but also it is required that a good pair of bearing metals occupy special positions in the Periodic Table of elements in order that the atomic junctions be weak and easily broken.

Wherever machine parts run in sliding contact with each other, it is desirable that they be made of materials that do not readily weld or otherwise "stick" together. Bearings should not weld to journals. Piston rings should not weld to cylinders. Gear teeth should not weld to meshing gear teeth.

Although it has been studied for many years, very little is actually known concerning the non-welding property of sliding surfaces, and what is known is largely empirical and frequently contradictory. In the absence of any general theory to guide development, bearing materials have evolved by cut-and-try methods.

The entire subject of sliding surfaces is of such obvious importance that recently it was decided to study it anew from the viewpoint of solid-state physics in the hope of gaining greater insight into the underlying phenomena. The results of this study conducted by Research Laboratories Division clarify and unify much of what heretofore has been known only on an empirical basis and provide a broad foundation for future developments in this field.

Bearing Experiments with Elemental Metals

If one seeks to understand the behavior of metals in sliding contact, one should have before him reliable data on how metals actually behave. There is no shortage of such data, but most deal with commercial bearing alloys that are too complex for theoretical study. For this purpose, it was desirable to have data on pure elemental metals whose physical and chemical properties are well known.

Of the 92 naturally occurring elements, over sixty may be classed as metals. Some—such as radium, polonium, protactinium, and virtually the entire series of rare earths—are so excessively rare that it is not feasible to test them as bearings. Others—mercury, gallium, sodium, and potassium—are readily available but have physical properties that are not compatible with the testing procedure. Consequently, the number of elemental metals suitable for testing as bearings is of the order of 40. Those tested ranged from lithium (atomic number 3) through uranium (atomic number 92) (Fig. 1).

Table I summarizes the test results. There are four ratings—*good*, *fair*, *poor*, and *very poor*—according to how well the bearings performed against steel when subjected to a series of increasingly heavy loads.

Following these tests a question arose whether the results had been influenced by the choice of lubricant. To investigate this question two additional series of tests were run—the first with water as the lubricant, the other (in a different apparatus) with no lubricant at all. These tests showed that the relative ratings of metals are not affected by the choice of lubricant. All metals fail at lighter loads when water is the lubricant and at still lighter loads when run absolutely dry, but the order in which they fail remains unchanged.

Atomic Contacts between Metals

When two metallic bodies are brought together, the attractive forces between matching surface atoms cause the bodies to adhere or weld together. The welding of one metal surface to another is achieved with metal surfaces that are free from contaminating films such as oxides, water, and others, depending on the history of the metals. It is not generally possible, therefore, to cause welding between metals by bringing them together, except in carefully controlled laboratory experiments where bare metal surfaces are brought together in a vacuum or reducing atmosphere. However, when metal surfaces slide against each other under load (such as when an oil film bearing starts up), the surface asperities of the metals must be deformed rapidly to allow sliding to take place. The resulting high local temperatures and pressures permit the removal of surface films from the asperities and the result is that there are areas of metal-to-metal contact between the bearing and the journal. Under such conditions poor bearing metals seize; good bearing metals survive. Thus, the problem is reduced to a consideration of what happens when bare metal slides on bare metal. This is the basis of the following discussion. Whether a pair of metals slide smoothly upon each other or seize violently together depends on two factors:

- How many atoms on the surface of one body match up with corresponding atoms on the surface of the other body.
- How strongly these atomic junctions adhere together.

It is not difficult to form a rough estimate of the number of atomic junctions

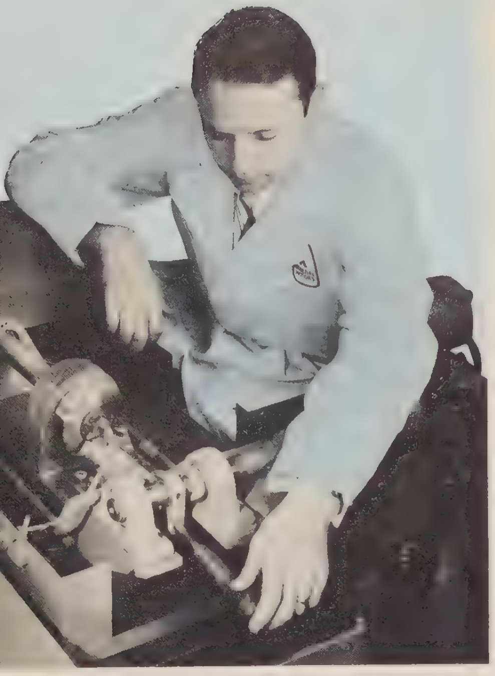


Fig. 1—Specimens of 40 elements ranging from lithium to uranium were obtained and fashioned into small slider bearings (5/8 in. square) for testing in this machine. In the test, the bearings were pressed at an angle of 1/2° against the surface of a metal disc rotating at high speed. Lubrication was provided by a stream of kerosene directed on the disc.

ELEMENTS TESTED IN SLIDING CONTACT WITH IRON

Atomic Number	Element	Atomic Diameter	Periodic Group	Bearing Rating
3	Lithium	3.03	I	Poor
4	Beryllium	2.22	II	Very Poor
6	Carbon	1.54	IV	Fair
12	Magnesium	3.20	II	Poor
13	Aluminum	2.86	III	Poor
14	Silicon	2.34	IV	Very Poor
20	Calcium	3.93	IIa	Very Poor
22	Titanium	2.91	IVa	Very Poor
24	Chromium	2.49	VIa	Very Poor
25	Manganese	2.66	VIIa	Very Poor
26	Iron	2.48	VIII	Very Poor
27	Cobalt	2.50	VIII	Very Poor
28	Nickel	2.50	VIII	Very Poor
29	Copper	2.55	Ib	Fair—Poor
30	Zinc	2.66	IIb	Poor
32	Germanium	2.70	IVb	Good
34	Selenium	2.90	VIb	Fair
40	Zirconium	3.16	IVa	Very Poor
41	Columbium	2.85	Va	Very Poor
42	Molybdenum	2.72	VIa	Very Poor
45	Rhodium	2.68	VIII	Very Poor
46	Palladium	2.75	VIII	Very Poor
47	Silver	2.88	Ib	Good
48	Cadmium	2.97	IIb	Good—Fair
49	Indium	3.24	IIIb	Good
50	Tin	2.80	IVb	Good
51	Antimony	2.90	Vb	Good
52	Tellurium	3.22	VIb	Fair
56	Barium	4.34	IIa	Poor
58	Cerium	3.63	IIIa	Very Poor
73	Tantalum	2.85	Va	Very Poor
74	Tungsten	2.74	VIa	Poor
77	Iridium	2.71	VIII	Very Poor
78	Platinum	2.77	VIII	Very Poor
79	Gold	2.78	Ib	Very Poor
81	Thallium	3.40	IIIb	Good
82	Lead	3.49	IVb	Good
83	Bismuth	3.10	Vb	Good
90	Thorium	3.60	IVa	Very Poor
92	Uranium	2.84	VIa	Very Poor

Table I—Bearing tests run on the elements listed in this table indicated that those elements which make good bearing materials invariably fall into the b-subgroup of the Periodic Table of elements (Fig. 2).

that may be formed between sliding metals. From X-ray diffraction studies it is known that all metals are *crystalline*—they are composed of an enormously large number of atoms arranged in orderly patterns which may be visualized as assemblages of parallelepipeds. The atoms arrange themselves in these orderly patterns to minimize the potential energy of the assemblage.

If two identical crystals (of copper, for example) are brought together in such a way that their axes are perfectly aligned, then for every atom on the surface of one crystal there will correspond a similar atom on the surface of the other crystal. The attractive force fields associated with these surface atoms will be perfectly satisfied, and the two surfaces will adhere strongly together. In fact, the atoms along the interface of the two crystals will be indistinguishable from those in the interior of the crystals, and the junction between the two crystals will be just as strong as the crystals themselves.

Next, consider what happens when two dissimilar crystals (of copper and lead, for example) are brought together in which the atoms are differently spaced as a consequence of their different sizes. Now, matching is not possible between all the surface atoms of one crystal and those of the other crystal. A few atoms will match perfectly. But most will be more or less mismatched in such a way as to give imperfect junctions.

These considerations suggest that if it is desired to minimize the adhesion between a pair of sliding metal surfaces, metals should be chosen whose crystal lattices are so different that there will be very poor matching between atoms on the contacting surfaces. In other words, choose metals whose atoms differ greatly in size.

When the test data of Table I are examined from this viewpoint, a remarkable agreement is found between what the theory would lead one to expect and what the experiment actually gives. The good metals are mostly those whose atomic diameters are at least 15 per cent greater than that of the iron atom (2.48 Angstrom units).* Thus, silver, cadmium, indium, antimony, thallium, lead, and bismuth all have atomic diameters of 2.68 A or greater.

It should be noted that some metals having atomic diameters of 2.86 A or greater do not rate good in the bearing tests. These include lithium, magnesium, calcium, and barium. At the same time, two metals rating good have atomic diameters less than 2.86 A. These are germanium (2.44 A) and tin (2.80 A).

---

*An Angstrom unit (abbreviated A) is 10^{-10} meter. It is the unit generally used in measuring atomic dimensions.

Strength of Atomic Junctions

The ability of a pair of metals to slide upon each other without welding together depends not only on the number of atomic junctions that may be formed between them but also on the quality of the junctions. A pair of metals that forms weak junctions in large numbers may be more desirable than a pair that forms smaller numbers of stronger junctions. In the first case, shearing is confined to the immediate interface, and there is little or no damage in depth. But in the second case, there is a great tendency for the junctions to wipe and smear, for pieces to be plucked from one or the other of the surfaces, and for damage to occur on a gross scale. The ideal is, of course, a pair of metals that gives few junctions and those junctions of a fragile sort.

What determines whether a junction will be strong or weak?

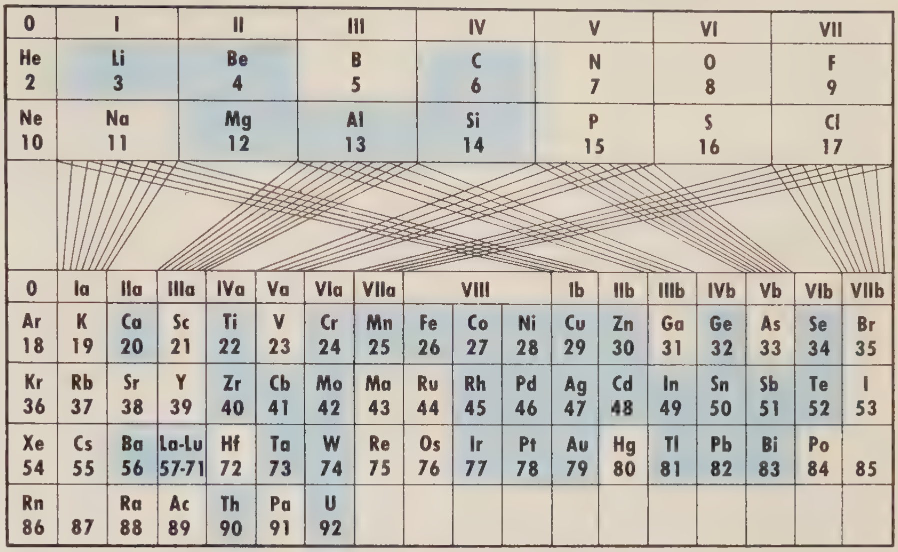
To answer this question it is necessary to examine the nature of the binding forces between atoms in solid metals.

There are two principal types of bonds. The first and more important is the *metallic bond* resulting from the attraction between positively charged metal cations and their associated negatively charged valence electrons. An important aspect of this type of bond is that the valence electrons are free to move about in the interstices between the cations; it is this mobility of the valence electrons that makes metals good conductors of electricity and heat and gives them other distinctive metallic properties. What is most important in this case, however, is that the metallic bond is a very tenacious bond. When pulled on, it gives but does not let go.

The other type of bond in metals is the *covalent bond* that arises from the sharing of valence electrons between ajacent atoms. The electron sharing is done in such a way that each of the participating atoms, in effect, completes its valence shell. The result is a very stable bond that is relatively rigid and unyielding. Consequently, substances in which the binding is purely covalent tend to be extremely brittle and fragile. Metals may combine to form intermetallic compounds in which the binding is wholly covalent.

If two metals form junctions that are wholly metallic, it should be expected that they would be an inferior pair of bearing metals—even though the number of junctions might be small. If two metals form junctions that are partly covalent, it should be expected that they be somewhat better bearing metals; how much better would, of course, depend upon whether there were many or few junctions. Finally, if two metals form junctions that are wholly covalent, it should be expected that they perform extremely well as bearing metals—even though the number of junctions might be very large.

THE PERIODIC SYSTEM OF THE ELEMENTS



0	I	II	III	IV	V	VI	VII
He 2	Li 3	Be 4	B 5	C 6	N 7	O 8	F 9
Ne 10	Na 11	Mg 12	Al 13	Si 14	P 15	S 16	Cl 17

0	Ia	IIa	IIIa	IVa	Va	VIa	VIIa	VIII			Ib	IIb	IIIb	IVb	Vb	VIb	VIIb
Ar 18	K 19	Ca 20	Sc 21	Ti 22	V 23	Cr 24	Mn 25	Fe 26	Co 27	Ni 28	Cu 29	Zn 30	Ga 31	Ge 32	As 33	Se 34	Br 35
Kr 36	Rb 37	Sr 38	Y 39	Zr 40	Cb 41	Mo 42	Ma 43	Ru 44	Rh 45	Pd 46	Ag 47	Cd 48	In 49	Sn 50	Sb 51	Te 52	I 53
Xe 54	Cs 55	Ba 56	La-Lu 57-71	Hf 72	Ta 73	W 74	Re 75	Os 76	Ir 77	Pt 78	Au 79	Hg 80	Tl 81	Pb 82	Bi 83	Po 84	85
Rn 86	87	Ra 88	Ac 89	Th 90	Pa 91	U 92											

Fig. 2—The 40 elements tested as slider bearings against steel are indicated by the shaded portions in the Periodic Table. Those elements which make good bearing materials are found in the b-subgroups.

The Periodic Table of elements may be used as a rough indication of the type of junction that will be formed between metals (Fig. 2). Elements in columns Ia through VIIa and VIII tend to form junctions with iron that are wholly metallic. The atoms of these metals are characterized by relatively empty valence shells and incomplete *transition shells* (those just inside the valence shell). Elements in columns Ib through VIIb tend to form junctions in which metallic binding and covalent binding are mixed together. These metals are characterized by fuller valence shells and complete transition shells.

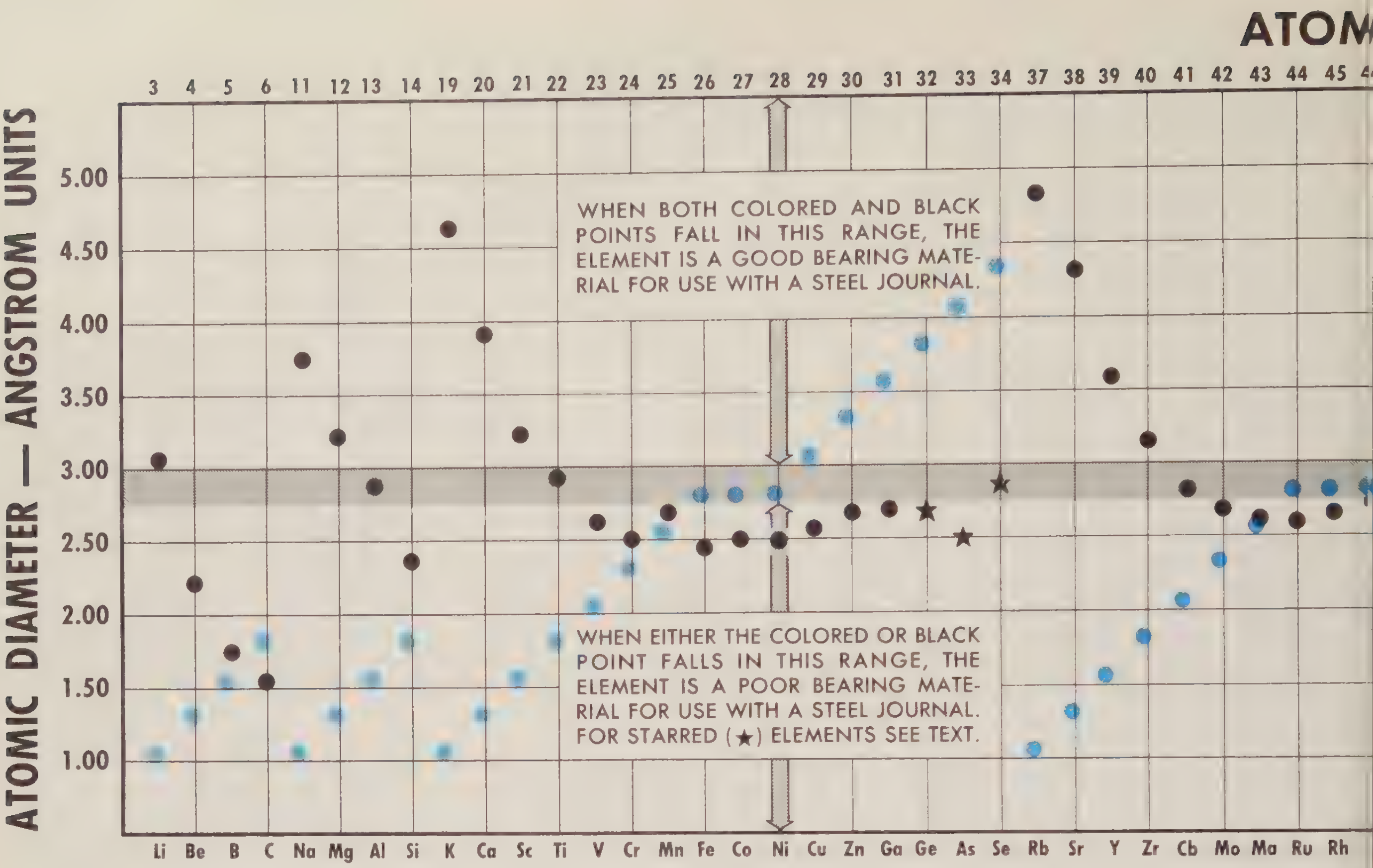
From these considerations it would be expected that the a-subgroup elements (those in columns Ia through VIIa and VIII) should make poor bearings when run against iron or steel. The b-subgroup elements (columns Ib through VIIb) should make better bearings. This is exactly what was found when the results of the bearing tests of various elements (Table I) were compared with their positions in the Periodic Table. The good bearing metals are from the b-subgroups.

Theory of Bearing Metals

Most of the good bearing metals satisfy two conditions—they have atomic diameters at least 15 per cent greater than iron and they occupy positions in the b-subgroups of the Periodic Table. The first condition implies a small number of atomic junctions between sliding metals. The second implies that the junctions are easily broken.

Certain metals—lithium, magnesium, calcium, barium—satisfy the first condition but are not good bearing metals. They fail to satisfy the condition with regard to position in the Periodic Table. Certain other metals—copper, zinc, gold—satisfy the second condition but are not good bearing metals. They fail to satisfy the condition with regard to atomic diameter. What is necessary is that both conditions be satisfied. When a metal satisfies both conditions (when it has both favorable atomic diameter and favorable position in the Periodic Table), then it is invariably a good bearing metal for use with iron and steel. This is true without exception (Fig. 3).

The converse is not necessarily true. There are two good bearing metals that do not satisfy both conditions. Germanium and tin are b-subgroup metals but



WHEN BOTH COLORED AND BLACK POINTS FALL IN THIS RANGE, THE ELEMENT IS A GOOD BEARING MATERIAL FOR USE WITH A STEEL JOURNAL.

WHEN EITHER THE COLORED OR BLACK POINT FALLS IN THIS RANGE, THE ELEMENT IS A POOR BEARING MATERIAL FOR USE WITH A STEEL JOURNAL. FOR STARRED (★) ELEMENTS SEE TEXT.

Fig. 3—This chart graphically portrays the relationship of the metallic elements to iron as being good or bad bearing materials. Those elements which have an atomic diameter approximately fifteen per cent or greater than that of iron and which appear in the b-subgroups of the Periodic Table are good bearing materials. The shaded band which runs across the middle of the chart indicates an area roughly fifteen percent greater than the atomic diameter of iron.

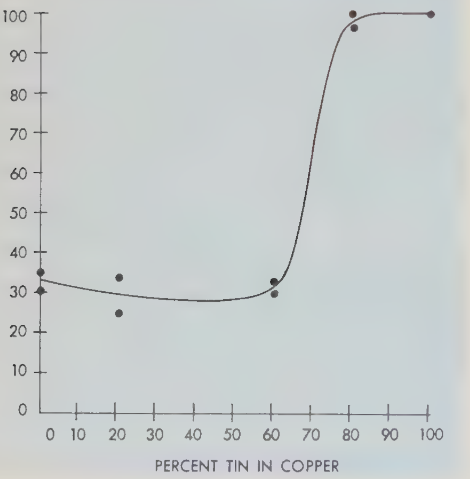


Fig. 4—Test results show that bronze bearings composed of the elements copper and tin have poor score resistance characteristics until the amount of tin in the alloy reaches 60 per cent or more.

have atomic diameters not greatly different from that of iron. The special characteristic of these metals is that they react with iron to form intermetallic compounds of definite stoichiometric composition with covalent binding. The resulting junctions are so brittle and easily broken that it practically does not matter whether there are many or few of them.

Other elements forming intermetallic compounds with iron are antimony, selenium, and tellurium. Antimony rates good as a bearing metal, while selenium and tellurium both rate fair. These elements forming intermetallic compounds with iron are denoted by a star in Fig. 3.

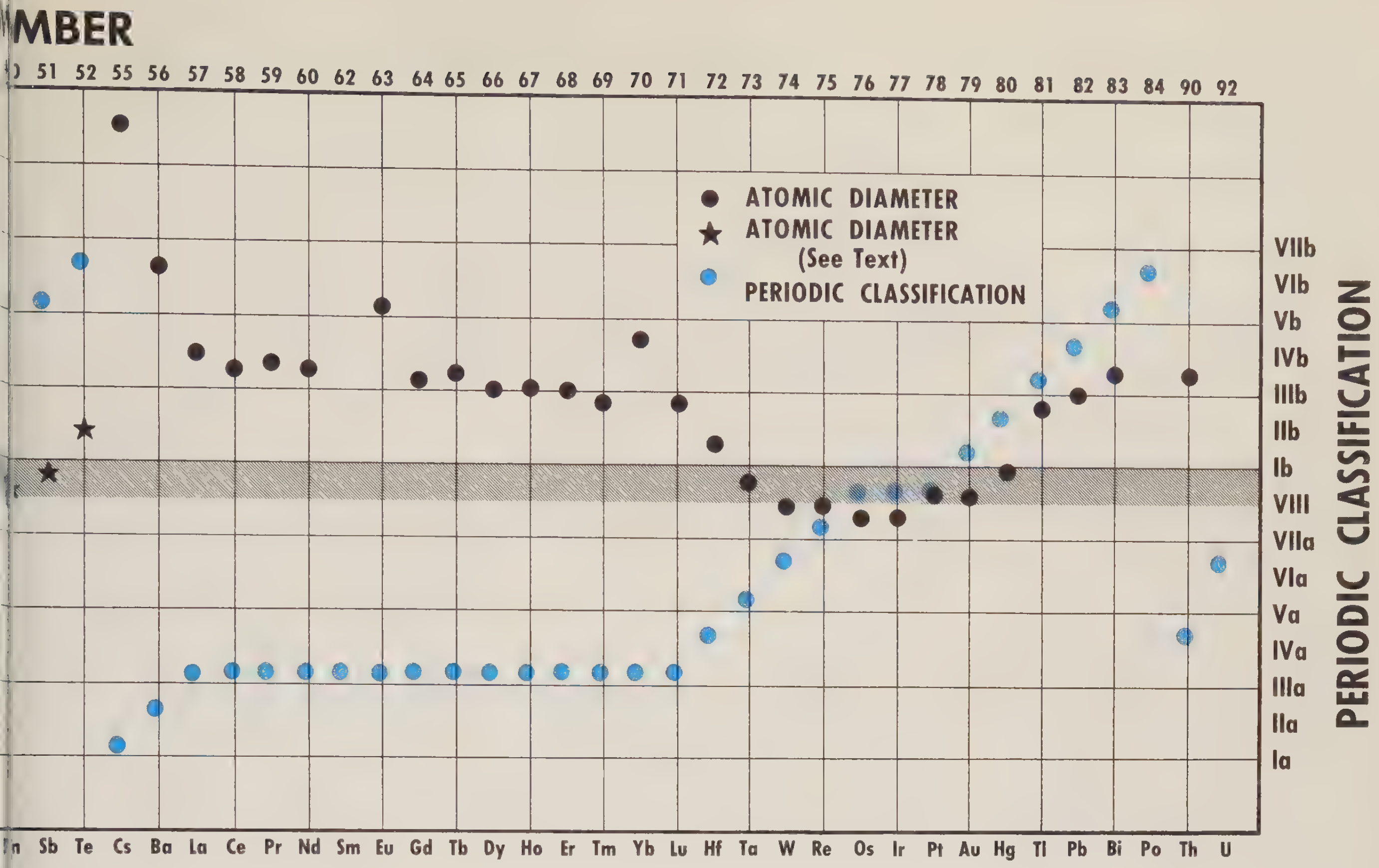
The two conditions of favorable atomic size and favorable positions in the Periodic Table (with the proviso just noted concerning intermetallic compounds) together constitute a complete theory of bearing metals to which there are no known exceptions.

Practical Applications

Only a few elemental metals find direct application as bearing materials. These are silver, lead, tin, and (to a lesser extent) indium and cadmium. With the possible exception of silver, these are extremely weak materials that must be used in thin layers reinforced by a strong backing to prevent their extruding under load. It may be noted that they all satisfy the conditions for good bearing materials.

A number of commercial bearing materials are composed of metals having relatively low solid solubility. Examples are copper-lead, aluminum-tin, aluminum-cadmium, and bearings of grided construction. Such materials are attempts to combine the superior bearing properties of metals like lead, tin, and cadmium with the superior strength of metals like copper and aluminum. Their distinguishing characteristic is a well-defined two-phase structure. In copper-lead, aluminum-tin, and aluminum-cadmium this two-phase structure results from the relatively low solubility of the constituent metals in the solid state. In grided bearings, it is produced by mechanical means.

For such two-phase materials to be successful two things are necessary. First, the stronger, higher-melting phase must carry the load. This is accomplished either by making it continuous, as with the aluminum matrix in aluminum-tin and



T

aluminum-cadmium, or by making it columnar, as with the copper dendrites in copper-lead. Second, the weaker, lower-melting phase must be one of the good bearing metals. This insures that in the event of distress the good metal will melt out first, smear over the bearing surface, and thus serve as a prophylactic to prevent contact between the journal and the stronger phase.

Of the various alloys used as bearing materials, many are simple solutions of two or more good metals. Examples are lead-tin alloy and lead-indium alloy. Such alloys are always good bearing materials. It is worth noting, however, that while alloying two good metals does not harm their bearing characteristics, it does not improve them either. Consequently, the objective in alloying such metals is to improve their corrosion resistance (unalloyed lead, for example, is attacked by oxidized oils), their castability, or some other property aside from bearing characteristics.

Only a few commercial bearing materials are alloys of a good metal and a metal of inferior bearing characteristics. An example is copper-tin bronze. Here the good constituent (tin) confers none of its desirable bearing characteristics on the alloy until it is present in amounts of 60 per cent or more (Fig. 4). Indeed, there is some evidence that the performance of copper as a bearing material is slightly depreciated by small amounts of tin. If correct (and it appears to be borne out by the data of Fig 4), this might be due to the effect of the tin in depressing the melting point of the solution.

Babbitt is still one of the most important of all commercial bearing materials. *Babbitt* is an alloy comprising a two-phase material consisting of hard particles in a soft matrix. In tin-base babbitt, for example, the constituents tin, antimony, and copper form a two-phase system having hard crystals of antimony-tin and tin-copper intermetallic compounds supported in or by a soft matrix consisting of a solid solution of tin and antimony. The hard particles stiffen the soft tin-antimony alloy in the same way that chunks of fruit stiffen a bowl of soft gelatin dessert, while the tin-antimony alloy (being composed entirely of good metals) retains the good bearing characteristics of its constituents.

Contamination by traces of inferior metals usually has an adverse effect on bearing materials. Very small amounts of zinc, for example, deprive babbitt of its good performance. In general, if the contaminant is a metal that goes into solution with the bearing material, its effect is harmful. But if the contaminant can be precipitated out of solution by suitable heat treatment, it will no longer have a harmful effect—provided that it has a higher melting point than the bearing material. If the contaminant has a lower melting point, there will, of course, be no object in precipitating it out of solution.

Summary

These examples show how the theory for elemental metals, properly interpreted, may be applied to the more complex materials used in commercial bearings. They show that the bearing materials that have successfully survived the test of time are those in which elemental metals of favorable atomic diameter and favorable position in the Periodic Table have been employed most effectively. The same general principles will serve as a basis for the selection of new bearing materials in new engineering situations.

Applying Material Handling Principles to Containers in the Manufacture of Appliance Parts

By **EDWARD L. BEERS**
Delco Appliance
Division

Ten rules for containers;
they work for small motor
laminations or furnace domes

Good manufacturing practice begins not only in the area of improved production techniques but also at the point where component parts are handled before assembly operations begin. To continue improving its manufacturing operations, General Motors long has devoted attention to material handling. The result has been the establishment of certain basic material handling principles which have evolved through the cooperation of various GM organizations. A number of these principles have been applied to problems at the Delco Appliance Division relating to the use of containers for handling motor laminations, furnace parts, and miscellaneous castings. Among the significant results obtained were reduction in container inventory, reduction in number of daily moves, increased storage capacity, improved methods, and reduced costs.

In the growing complexity of modern technology, engineering discipline needs to be applied to every conceivable area in order to achieve cost reduction and utmost efficiency. One such productive area is that of material handling where, in recent years, basic principles have been established in manual form for future guidance.[1] Through the firm application of these basic principles, Delco Appliance Division has succeeded in improving the handling of armature and core laminations for the high-volume production of small fractional horsepower motors used in a number of automotive applications.

A problem in material handling is treated with the same basic engineering approach as a problem in other fields of engineering. This basic step-by-step approach is as follows:

- Determine both the immediate and the long-range problem, or objective.
- Study the conditions causing the problem.
- Plan the possible solutions in the light of established material handling principles.
- Evaluate the possible solutions in terms of plant, manpower, facilities, materials, and the product.
- Recommend action; that is, sell the plan to management.
- Follow up to assure effectiveness; that is, educate the operating personnel.
- Check the results of the new program to see that the proposed plan is working effectively.

A study of the fourth item listed will show that the detailed solution of similar material handling problems may vary because of the different characteristics present in manufacturing plants. These characteristics may result from the type of plant construction, existing equipment, items produced, manpower used, or from many other sources. However, invariably the same basic principles can be applied to the problem, thus controlling the course of action along those lines which ultimately lead to more effective handling.

A study of several examples at Delco Appliance will show how 10 of the basic material handling principles were applied to container problems. These examples related to material handling problems are classified into four general groups as follows:

- Containers for handling motor laminations
- Containers for handling incoming materials
- Economics of containers
- Follow-up.

What Is a Container?

A definition of a *container* as it is used in the following examples may be stated as "a means which provides the necessary enclosure using compatible material arranged so as to properly retain the product and restrain its movement to the degree necessary for protection, handling, and storage."[1]

A study of this definition shows that the following equipment falls into the container classification:

- Pans
- Tubs, baskets (collapsible or rigid)
- Wooden crates (returnable or expendable)
- Pallets and skids (returnable or expendable)
- Tubular steel racks.

Containers for Handling Motor Laminations

New containers and handling methods were developed for handling motor laminations based on the following two basic principles:

- "*Containers must provide necessary enclosure to retain the product properly and restrain its movement to the degree necessary for protection, handling, and storage.*
- "*Whenever possible, containers should be standardized.*"[1]

A study of the former method of handling laminations indicated that the essential problems to be solved were: (a) reduction of container inventory, (b) improvement of stacking or storage, and (c) improved protection of parts while in containers.

Three types of laminations were handled, namely heater motor armature laminations, heater motor field laminations, and split-phase rotor laminations. An inventory of 2,500 pans and boxes, 100 skids, 20 trucks, and 300 intermediate pallets was required formerly to handle these laminations, and no provision was included for stacking of unit loads. The number of daily moves required for various containers was estimated at 3,100, exclusive of moves in a machine cycle. A substantial number of these moves involved the handling of wooden boxes and pans, both full and empty.

Applying the Principles to the Problem

At the outset, it was apparent that special containers would be needed. The problem affected two major products (automotive heater motors and split-phase motors for various appliances) and, therefore, the ultimate design had to be a container that would blend into the existing standard container system. This system was composed basically of the steel tub, the wire basket, two standard types of tote pans, and a tubular steel coil rack. At this point, a container choice had to be made, guided by the application of the two aforementioned principles. The need for container standardization and the storage and handling conditions of the work area all pointed to the desirability of an enclosed container accessible on one side having one-half the capacity of the steel tub.

Results

The result was a new lamination rack built to the same perimeter dimensions as the existing steel coil racks—33 in. by 19 in.—consisting of a bottom and three sides (Fig. 1). This container had a unit load capacity of 5,000 lb and could be moved by all types of existing plant handling equipment. In addition, it could be stacked interchangeably with the coil racks.

Another element of the basic principles to be considered was that of part protection. In the former system, pans had been used as containers which served also to retain the laminations on the spindles. The proposed rack design, however, did not utilize pans; thus, the spindles were studied for a new method of retaining the laminations. The solution was to insert spindles into both ends of the 18-in. long stacks of armature laminations (Fig. 1).

The use of a tray for handling field laminations was continued, since this proved basically effective because of the design of these laminations. The tray was merely re-designed to fit the racks (Fig. 2).

In the case of split-phase rotor laminations, the retaining feature was accomplished by wire pins which were inserted into transverse holes drilled in the ends of the U-shaped spindles (Fig. 3). Fortunately, inserting these pins did not necessarily mean increasing the production cycle time as this operation took advantage of operator idle time within the machine cycle. These pins were handled in a dead soft condition.

Comparative Analysis of Old and New Methods of Handling Laminations

A comparison between the old and new methods of handling laminations may be summarized as follows:

- A reduction in the container inventory from approximately 2,900 pans to 120 racks was realized. In terms of daily moves, this meant a reduction from 3,100 to 1,780 items.
- A 950 per cent increase in the capacity to store armature laminations was achieved.
- A 350 per cent increase in the capacity to store split-phase rotor and heater-motor field laminations was achieved.
- A positive system of handling from the standpoint of part damage and safety was instituted.

The results achieved in these examples illustrate that, no matter how difficult a material handling problem may seem in the beginning, it is imperative that the ultimate design or system be developed around sound principles. Should the basic principles be scrapped in order to ease the design problem, the result often may lead to an unsound and uneconomical system.

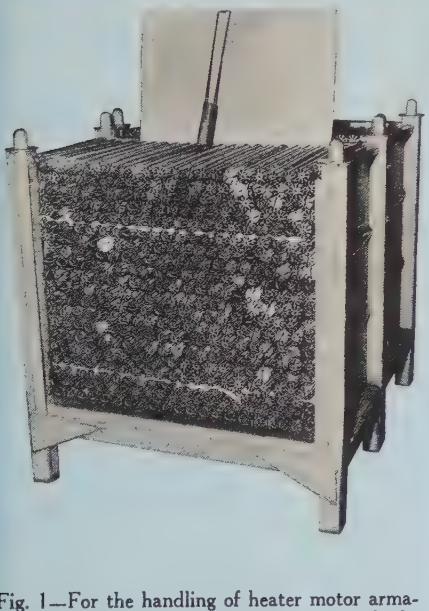


Fig. 1—For the handling of heater motor armature laminations, racks of this design were built. The rack has a capacity of 5,000 lb and was built to the same outside dimensions as other racks already in use. This permitted handling with existing equipment and stacking interchangeably. Shown at the top of the rack is the method of inserting spindles through individual stacks of laminations. Spindles are inserted in both ends to retain the laminations on the spindles.

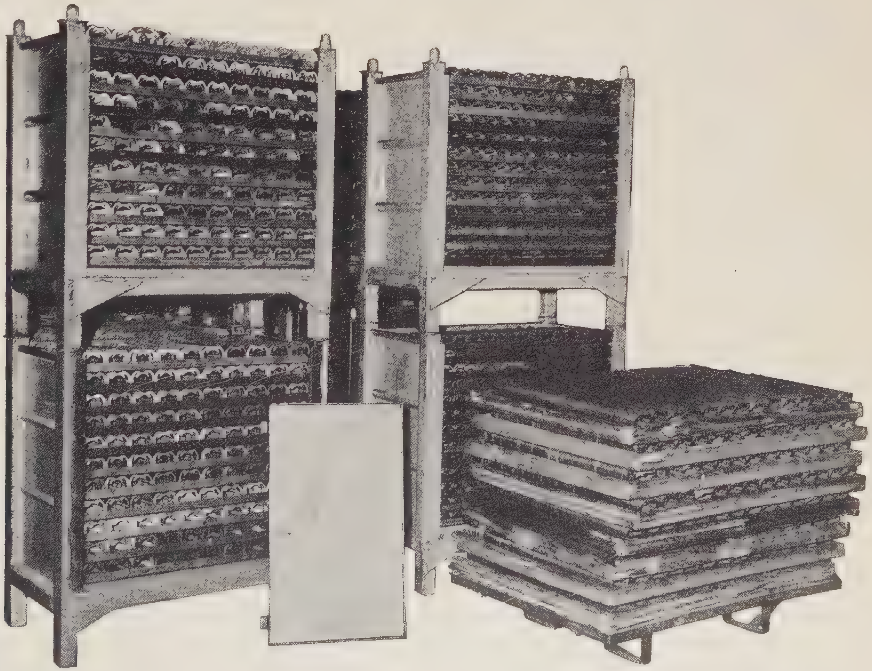


Fig. 2—Heater motor field laminations are handled on trays. Under the new system, the trays were re-designed to fit new steel racks, thus allowing the stacking feature. The simple tray design is shown in the foreground.

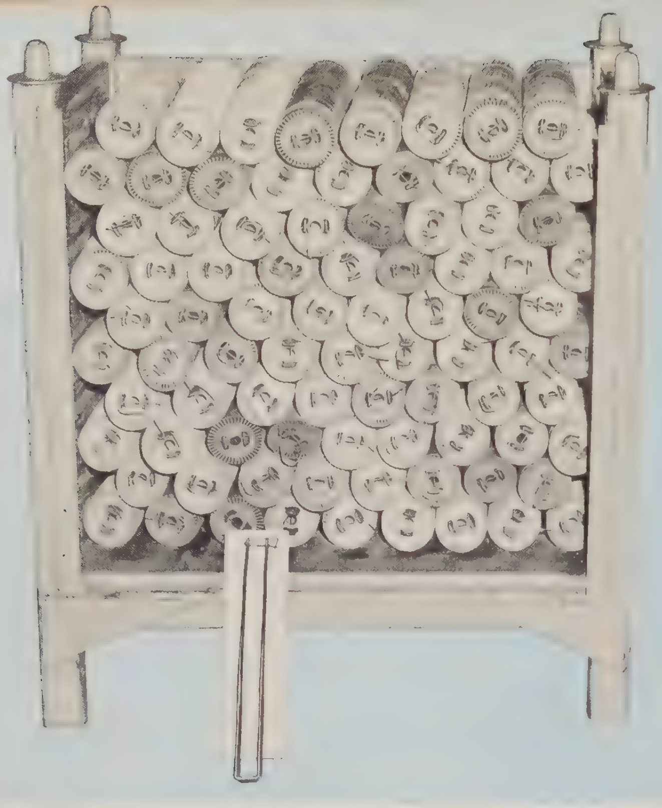


Fig. 3—Split-phase rotor laminations are handled in stacks which are retained by U-shaped spindles. This illustration shows a typical load of these laminations in a newly designed rack. Shown in the foreground is a spindle illustrating the use of a soft wire pin which is inserted through holes in the end of the spindle to retain the stack.

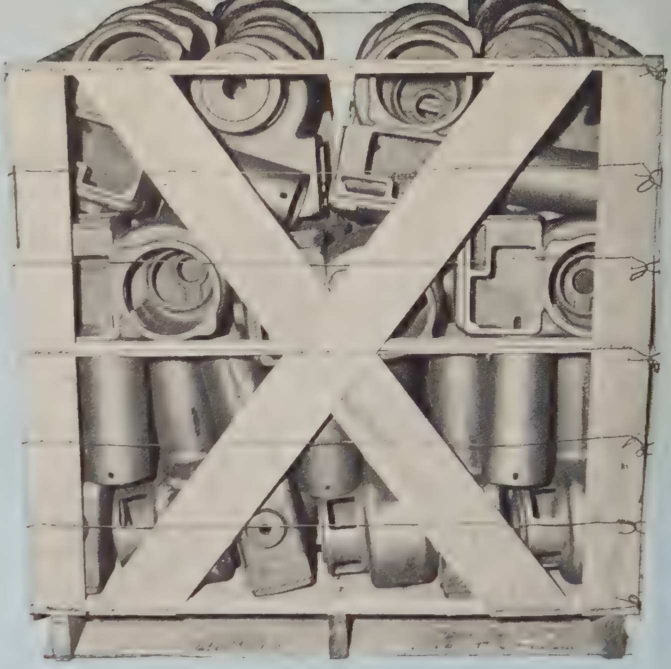


Fig. 4—The application of basic material handling principles resulted in this container design for the handling of castings for oil burners. The container is expendable, is furnished by the supplier, can be handled either by rail or truck shipments, and affords protection to the parts.

Containers for Handling Incoming Materials

Four additional principles of material handling were applied to a number of problems affecting incoming materials at Delco Appliance. These principles were:

- "*Containers should be designed for shipping and storage as well as for use at the point of use.*
- "*Containers should be designed of such size and construction as to permit their use in either rail car or highway truck and to make the most efficient use of floor space in either.*
- "*Expendable containers eliminate the cost of return and maintenance of returnable containers.*
- "*The weight of the container will determine how far it can be economically shipped and returned.*"[1]

The various problems occurring with incoming materials could be classified as follows:

(a) Periodic container shortages, brought on by inventory fluctuations

(b) Excessive handling devoted to receiving materials

(c) Ineffective utilization of cubic capacity in raw material stores

(d) Excessive scrap losses

(e) Safety hazards and poor housekeeping conditions.

To illustrate how these problems were solved, the handling of various types of castings will serve as an example, although the same problems applied generally to other incoming materials as well. The castings were of three types: (a) oil burner housings, (b) split-phase motor end bearings, and (c) miscellaneous castings.

Oil Burner Housings

Castings for domestic oil burner housings are relatively large and bulky and these were received loose—under the former handling methods—from at least three different suppliers. The castings were loaded into steel tubs 42 in. by 36 in. by 24 in. which provided a capacity for only 25 to 30 castings per tub. Thus, when a volume shipment of castings was received, it was sometimes difficult to avoid a number of problems which included (a) time consumed in locating a large number of tubs, (b) tying up the plant supply of tubs, and (c) storing of loose castings on the floor (if tubs were not available), introducing additional problems of storage space, casting breakage, and plant housekeeping.

Split-Phase End Bearings

The castings for split-phase motor end bearings are 6 in. in diameter and in the former system were received in a collapsible wire basket 46 in. by 36 in. by 24 in. having a capacity of approximately 350 pieces. The baskets, each weighing 145 lb, were furnished by Delco Appliance to the supplier. While this system had merit as a handling method, the problems of shipping expense covering empty baskets to the supplier and repairs to the baskets suggested that further improvement could be made.

Miscellaneous Castings

The miscellaneous group of castings discussed in this example consists of a variety of types, received usually in small volumes. These castings were received from as many as 10 different suppliers and were received loose or in burlap sacks, crates, or 55-gal drums. In the handling of these castings, it was seen that improvements could be made to solve the problems affecting (a) tub supply (where required), (b) different types of containers mixed into the plant system, and (c) the best use of available cubic storage space.

Principles

On the surface, an easy solution to the problem of handling these incoming castings appears to lie in the purchase of more tubs and baskets. This action would

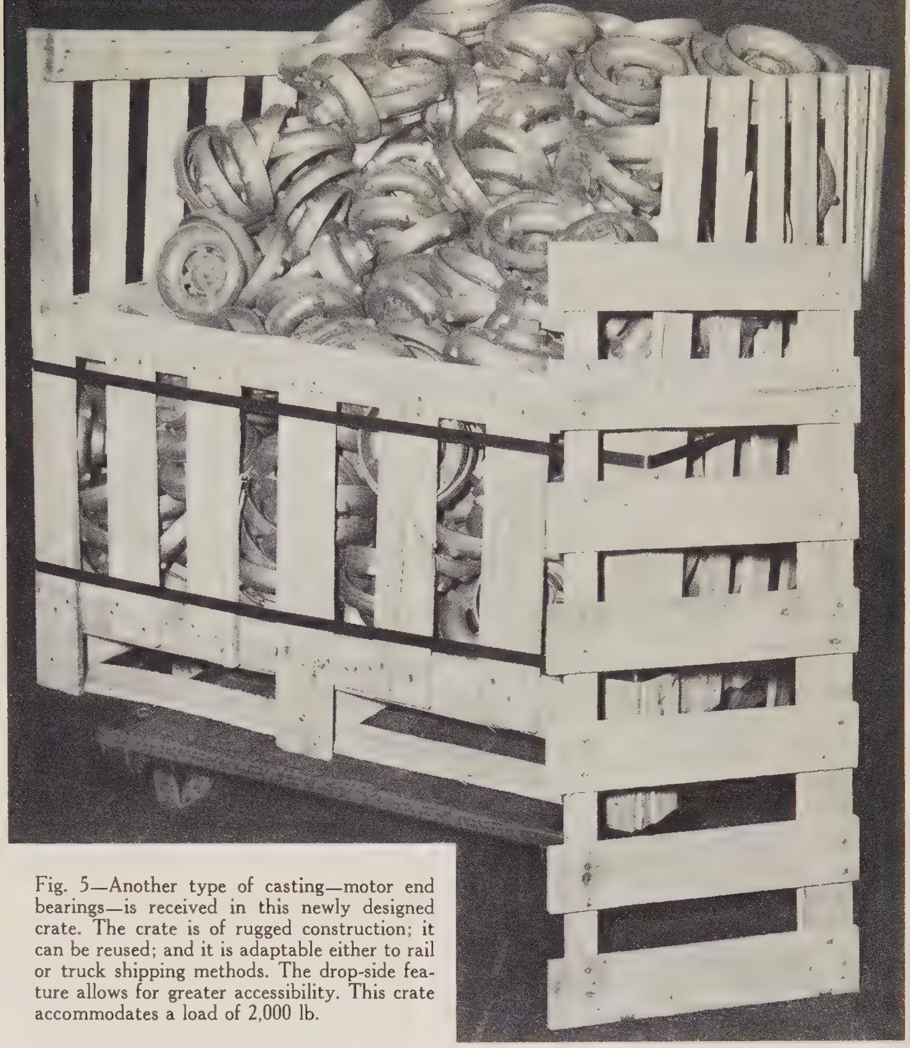


Fig. 5—Another type of casting—motor end bearings—is received in this newly designed crate. The crate is of rugged construction; it can be reused; and it is adaptable either to rail or truck shipping methods. The drop-side feature allows for greater accessibility. This crate accommodates a load of 2,000 lb.

be impractical, however, not only because of the cost of additional tubs and baskets but also because the basic material handling principles still would not be satisfied.

The solutions to the various problems described were found instead by studying and applying the four material handling principles affecting incoming materials, as listed earlier in this discussion.

In the case of handling oil burner housings—which continue to be received from at least three suppliers—an expendable, wire-bound, wooden crate was developed which satisfied the requirements of a container suitable for shipping, storage, and use in the plant and was adaptable to either rail or truck shipment (Fig. 4). Although the prorated cost per piece seemed relatively high, it was offset by the saving in prorated scrap and handling costs by the former method. In addition, the new container made shipment by rail possible, which resulted in further cost reduction and also allowed the plant receiving department to unload during that part of the day when incoming truck shipments were at a minimum.

Solving the problem of handling castings for motor end bearings required a cost comparison of the methods using the returnable wire basket and the expendable wooden crate. The results showed that the monthly returnable freight costs of the wire baskets would offset the monthly costs of expendable crates; also certain additional advantages would result:

(a) The system could be geared to production releases and the cost incorporated into the product since the crates would be furnished by the supplier.

(b) About one-half of the crates could be reused since they could be built of rugged construction to carry a load of 2,000 lb and also to stack in tiers. By designing the crates so that steel strapping was the sole means of holding the crates together, they could be knocked down and reused by local suppliers.

(c) In some cases, the service could be carried out on a "no charge" basis where a supplier might use salvageable crates and use his own trucks for delivery. This practice could offset those cases where prorating the crate costs over a number of pieces per crate would be uneconomical due to high crate costs as a result of small orders.

The resulting crate which was designed and put into use is shown in Fig. 5.

During the design of both crates, it was important to adhere to the two principles relating to point of use and overall size to meet the specifications required when shipping by rail cars and highway trucks. The end-bearing crate, for example, was designed with a drop-side feature for ease in unloading at the work place. However, due to net weights, the crate dimensions had to be limited to 40 in. by 40 in. which—while satisfactory for highway shipments—created lost space in rail shipments. Fortunately, both types of containers could be stacked which afforded full utilization of the cubic storage capacity in the plant.

As in the example of the handling of motor laminations, it again can be emphasized that by holding to the basic principles, the final solution will be a sound and an economical one. In reviewing the new system of handling incoming castings, it can be seen that the system is economical to operate, does not require additional equipment, and it released approximately 550 containers of the previous type for in-plant use.

Economics of Containers

The economics of container design and the overall investment required to apply the design are important factors in material handling. The following three principles aid in controlling container costs and the operating costs of the container handling systems:

- "*Unit loads should be increased to an economic maximum.*
- "*Inventory of containers should be kept to a planned minimum.*
- "*Minimum materials with maximum strength should be used in container construction.*"[1]

When the unit cost of the tub, wire basket, pallet, pan, and rack was studied with respect to overall plant inventories, the total investment relative to various handling systems became apparent. For

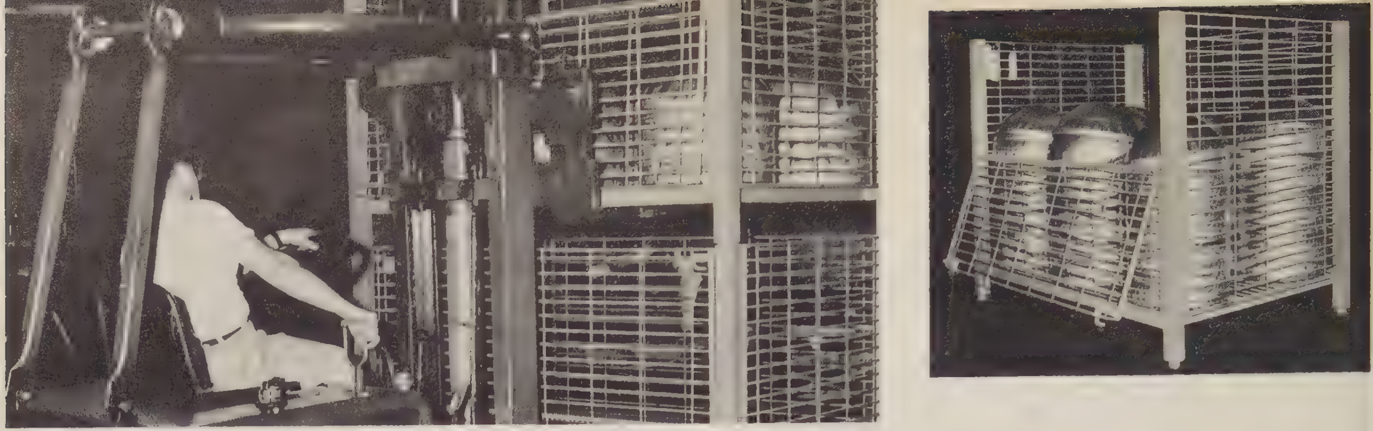


Fig. 6—This re-designed wire basket, 68 in. by 46 in. by 30 in., was adopted for the handling of furnace domes. (Left) It provides the stacking feature and increases the unit load capacity over the former baskets. (Right) The drop-side feature provides increased accessibility.

example, it was first suggested that special wire baskets, which were already in use, be used to handle some bulky parts. Although bulky, these parts could nest and stack within each other. Therefore, a study of the economics involved in the use of a special basket versus a system of standard wooden pallets and pallet racks was undertaken. Results of the study showed that the projected investment in baskets would be three times greater than that for wooden pallets and racks for the same number of parts.

Another example of container economics in applying the first principle listed above can be shown in the use of the same wire basket. The part to be handled was a 22-in. diameter dome for a home heating furnace. It was clear from a cost standpoint that the existing 42 in. by 36 in. by 24 in. wire baskets could not handle these parts economically; therefore, a special 68 in. by 46 in. by 30 in. basket was designed. The ensuing design increased the unit load over the former system used, but not enough to justify the unit cost of the proposed container. The answer lay in increasing the depth of the container while, at the same time, giving consideration to container accessibility at the work place. This was accomplished by increasing the depth to 48 in. and hinging one side so that it could be dropped (Fig. 6 right). As a result, container capacity was increased 60 per cent at only a slight increase in cost.

Again, factors may enter which vary between plants, or even within one plant, which would alter conditions to the extent that this particular solution would not be economically practical. For one plant, the use of a basket might prove economical while for another, the use of racks and pallets might be the answer. The solution to both problems might depend primarily on product design. Here again the decisions made should be founded on application of the sound principles which control the course of action toward an economical and effective system.

Follow-Up Period

A final principle was applied to the follow-up of the new system, namely:

- *The education of all personnel concerned with the new container or handling system is imperative.*

This principle of job follow-up is by far more important than the planning function which must precede it—for a plan is not carried out by the planners but by those people on the operational level. The success of the plan is measured by how well it is received by these people, and how stimulated they are to using it.

For instance, when the special wire baskets to handle furnace domes were first put into use, everyone directly and indirectly concerned with the proposed plan had to be notified. The plan was quite simple—only three different parts were to be handled in the baskets. This instruction had to be given to a large group of employes because, present in the same area, were smaller and heavier parts which if used in the same containers would result in container overloading. Supervision also played an important part in this program through direct instruction to the employe. However, it was also the responsibility of the engineer-in-charge to aid in this instructional phase and to make sure everyone was contacted and satisfied.

Educating personnel in the operation of the system is not the engineer's sole reason for participating in this instruction period. During this phase, the shortcomings which are invariably present in any new plan are uncovered and the engineer-in-charge can then make the adjustments or changes needed to insure more effective operation. Neglect of this phase can prevent the plan from functioning as efficiently as initially proposed. In the final analysis, the new plan must be operated properly to insure an improvement over the former methods.

Conclusion

The 10 principles discussed represent a few of a much larger group which can be applied to material handling as well as to problems in other fields of engineering. New principles are constantly being developed and added to the list.

By applying these principles at Delco Appliance, a number of improvements in the material handling system resulted which include: reduction of container inventory, reduction of the number of handlings per day, increase in storage capacity, improved plant safety conditions, reduction in receiving time, and reduced shipping costs on some items.

These examples demonstrate the concept of thinking in terms of basic principles and applying them to the gathered data which is recognized as the essential part of the formula for an economical and effective solution to engineering problems.

Bibliography

1. *Basic Reference Manual for Material Handling*, issued cooperatively by General Motors Divisions and General Motors Institute.

Other related literature in this field includes the following:

Berger, W. Walter, "A Design Study: Planning Maximum Storage and Shipping Ease and Economy into Commercial Vehicle Bodies," *General Motors Engineering Journal*, Vol. 1, No. 4 (January-February 1954), pp. 20-25.

The Theoretical and Practical Aspects of Automotive Brake Design and Testing

By FRANK V. KLCO
Oldsmobile
Division

One of the most immediate reactions, perhaps, to the subject of passenger car safety is the question "How good are the brakes?" This is only natural because the safety of the driver, passengers, pedestrians, other vehicles, and public property depends on the ability of the brake system to control and stop the motion of a vehicle. The overall design of a brake system, which is based on the concept of converting the kinetic energy of a moving vehicle into heat energy which, in turn, must be dissipated to the atmosphere, requires the engineer to consider many factors. Oftentimes, the theoretical and practical aspects of the design, from the drawing board stage until the final proving ground test, necessitate a compromise. The final design is one which the engineer believes possesses the optimum compromise of the factors required to provide the utmost in motoring safety.

A vehicle's kinetic energy is converted to heat energy and the heat then dissipated

An automotive brake is primarily a device designed to retard the motion of a vehicle by converting the moving vehicle's kinetic energy into heat energy. The braking device requires a rotating surface and a stationary surface which, when brought into contact, create a frictional drag which reduces the speed or stops the motion of the vehicle. The rotating surface is a brake drum mounted on each wheel of the vehicle. Inside each drum are two stationary brake shoe assemblies which, when forced outward, contact the inner surface of the brake drum. The brake shoes are mounted on a backing plate which, in turn, is mounted to a fixed structure such as the axle housing. Fastened to each brake shoe's outer surface is a friction material called a brake lining.

When the brake shoe lining is brought into contact with the inner surface of the rotating brake drum, the friction between these two members converts the drum's kinetic energy into heat energy. This heat is absorbed by the braking members and dissipated into the atmosphere. The brake drum serves as the primary heat exchanger and must be capable of absorbing the large quantities of heat produced on its inner surface and conducting this heat to the drum's exposed outer surface where it is radiated into the surrounding atmosphere.

Among the prime factors to consider in determining required braking performance are the gross weight and the maximum speed of a vehicle. These factors are important because the kinetic energy in a moving vehicle varies as its weight and the square of its speed, or

$$\textit{kinetic energy} = WV^2 / 2g \ (\textit{ft-lb})$$

where

W = gross weight of the vehicle (lb)
V = maximum velocity of the vehicle (ft per sec)
g = gravitational conversion factor (32.2 ft per sec^2).

The concept of kinetic energy points out that a heavier weight vehicle requires brakes having a greater kinetic energy conversion and heat dissipation capacity than a lighter weight vehicle. For example, an automobile having a gross weight of 4,500 lb and traveling at a speed of 70 mph (102.7 ft per sec) will possess $4{,}500(102.7)^2/64.4$ or 737,000 ft-lb of kinetic energy. This is equivalent to 947.3 Btu of heat which must be dissipated by the brakes in making one complete stop from a speed of 70 mph. An automobile having a gross weight of 3,500 lb, on the other hand, will require brakes capable of dissipating 736.8 Btu of heat when making a complete stop from a speed of 70 mph.

A theoretically sound brake design must be capable of providing dependable control of the vehicle under all normal driving conditions and should be able to withstand continuous use when descending long grades. The brakes also must be able to withstand rather severe braking in emergency stops without loss of vehicle control or braking ability.

Design Factors

Because automotive brakes must be capable of performing under varying speeds, loads, road conditions, and climatic conditions many factors must be taken into consideration in their design. Then the final brake design must be tested thoroughly in actual operation under these varying conditions to insure that the desired operational characteristics and performance have been obtained.

Load Distribution

The load (weight) distribution between the front and rear wheels is an important factor in brake design because the braking effort on each wheel is directly proportional to the effective weight on the wheel and the tire-to-road coefficient of friction. Two types of weight distribution must be considered by the designer—static and dynamic. *Static weight distribution* is the condition present when the vehicle is at rest. *Dynamic weight distribution* exists when the vehicle is in motion.

Fig. 1 shows two conditions of static weight distribution. For a vehicle at rest on a level surface, the braking force available is equal to the static weight on each wheel times the tire-to-road frictional force. When a vehicle is at rest on an incline, the effective weight normal to the road surface on each wheel becomes $W \cos \theta$, where θ is the angle of incline and W is the weight of the vehicle. The effective weight distribution is changed by the angle of incline, the relation of the height of the center of gravity to the wheelbase, and the horizontal distance between a plane through the center of

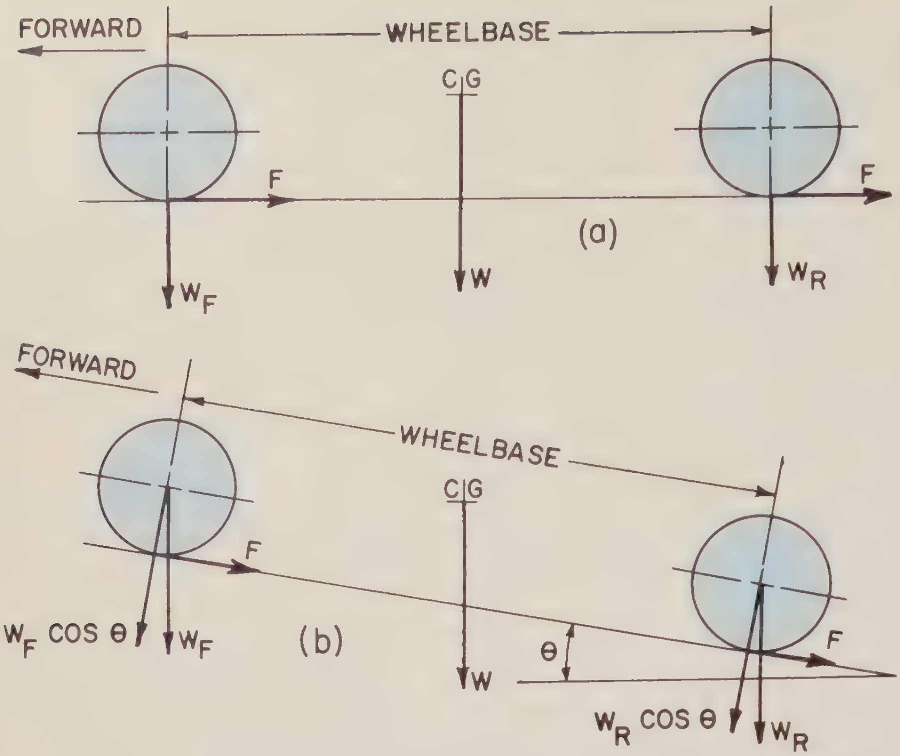


Fig. 1—The total weight W of a vehicle which is distributed between the front and rear wheels during braking is an important factor in brake design because the braking effort, or force, on each wheel is directly proportional to the effective weight on each wheel multiplied by the tire-to-road coefficient of friction. When a vehicle is at rest on a level surface (as shown in *a*), the braking force F available is equal to the static weight on each front wheel W_F and on each rear wheel W_R multiplied by the tire-to-road coefficient of friction. When a vehicle is at rest on an incline having an angle Θ (as shown in *b*), the effective weight normal to the road surface on each front wheel is $W_F \cos \Theta$ and on each rear wheel is $W_R \cos \Theta$.

gravity and a plane through the center of the axle.

The dynamic weight distribution is determined by the amount of weight transfer which, in turn, is dependent upon the deceleration rate and the height of the center of gravity. When the brakes are applied while the vehicle is in forward motion, a couple is produced which tends to rotate the vehicle about its center of gravity and transfer more effective weight to the front wheels.

Deceleration Requirements

The basic factor limiting braking performance that can be obtained on any vehicle is the coefficient of friction between the tire and the road surface. When a vehicle is braked, the maximum braking force obtainable is

$$F = \mu W$$

where

F = braking force (lb)
μ = tire-to-road coefficient of friction
W = total weight of the vehicle (lb).

The deceleration of a vehicle is directly proportional to its mass or

$$a = Fg/W$$

where

a = vehicular deceleration (ft per sec²)
F = braking force (lb)
W = total weight of the vehicle (lb)
g = gravitational conversion factor (32.2 ft per sec²).

By substituting μW for F into the above equation, the maximum deceleration obtainable can be made to be directly dependent upon the tire-to-road coefficient of friction or $a = \mu g$. When the velocity V and deceleration a of the vehicle are known, the time t in seconds and distance s in feet required for the vehicle to stop can be calculated from the basic formulae $t = V/a$ and $s = V^2/2a$.

Deceleration tests have shown that maximum controllable deceleration is obtainable at the *point of impending skidding* which is the point just before the wheels lock and begin to slide. Once the wheels cease to rotate and the tires begin to slide on the road surface, the tire-to-road coefficient of friction, and consequently the deceleration, decrease. The explanation for this lies in the fact that the friction between two surfaces at rest is greater than when they are in motion. The theory of friction states that when two bodies are in contact—in this case the tire and road surface—and have no relative motion with respect to each other, static friction occurs. When a force of increasing intensity is applied, in this case the braking force, the frictional resistance between the tire and the road surface increases in proportion up to the point of impending skidding. At this point, the applied braking force begins to exceed the frictional resistance available and the tire slides on the road surface with a resulting decrease in friction down to the kinetic value of friction between the tire and the road. The friction in the axle bearings and the propeller shaft and the rolling friction of the tires on the road surface combine with the static friction between the tire and the road, thus increasing the total frictional resistance available. Tests show that the coefficient of friction for impending skidding is from 10 per cent to 25 per cent or 30 per cent higher than that for straight skidding.

On dry concrete pavements, with tires in good condition, a coefficient of friction of approximately 0.70 can be reasonably expected at average driving speeds. This would give a maximum deceleration of $a = \mu g = 0.70(32.2) = 22.5$ ft per sec². Brakes are designed to obtain maximum deceleration. Normal brake stops, however, are made at rates below 10 ft per sec², which is within the range most passengers find comfortable.

Brake Effort Distribution

When the brakes are applied on a forward moving vehicle, the braking force acts along a tangent to the wheels at the point of ground contact. This braking force can be resolved into an equal and parallel force acting through the center of gravity of the vehicle and the previously mentioned couple tending to rotate the vehicle about its center of gravity. The moment of this couple is equal to the product of the braking force F times the height h of the center of gravity above the road surface.

The couple tending to produce rotation about the center of gravity is balanced by a couple which has a moment equal to the product of the amount of weight

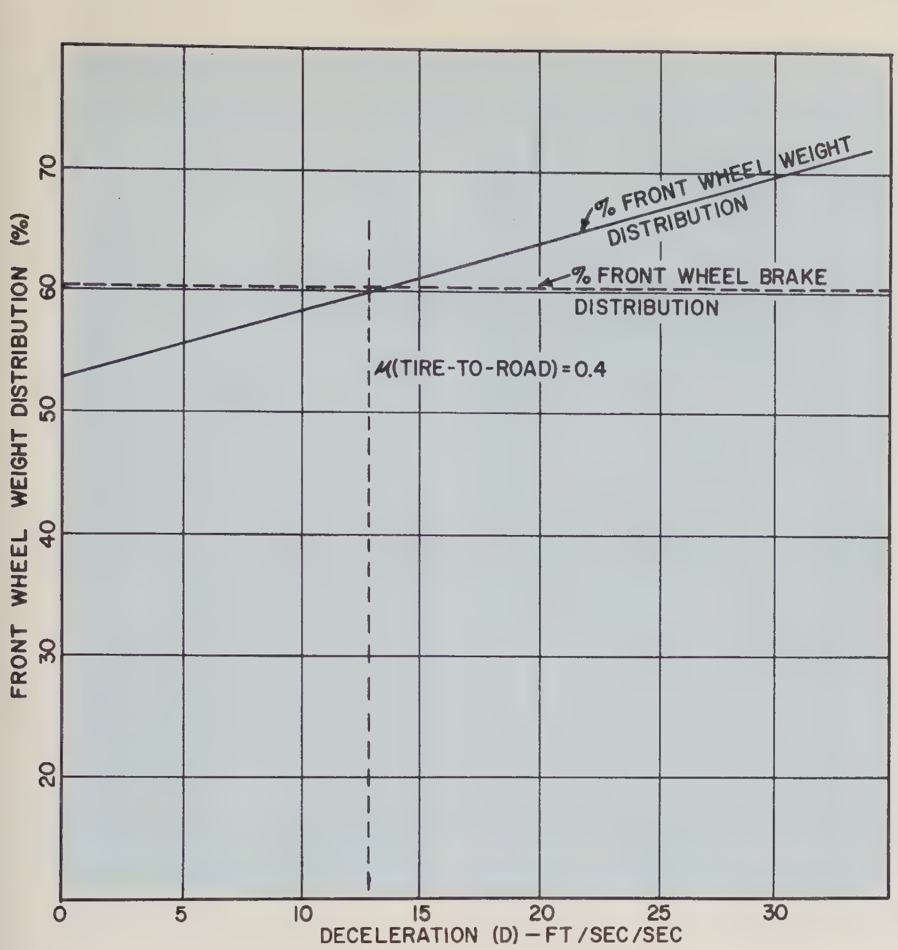


Fig. 2—The total weight of a vehicle distributed between the front and rear wheels varies with each change in rate of deceleration. When a vehicle is braked, the major portion of the vehicle's weight is distributed to the front wheels. This requires that the percentage of braking done by the front wheels be greater than for the rear wheels. The graph shows the percentage of front wheel braking effort at various deceleration rates and front wheel weight distributions. The deceleration rate chosen for this particular vehicle is 12.88 ft per sec^2 at a tire-to-road coefficient of friction of 0.4. This gives a weight distribution of 60.1 per cent of the total vehicle weight on the front wheels. The brake, therefore, would be designed so that the front wheel braking effort is approximately sixty per cent of the total braking effort. The graph is plotted from the formula $W_F = W/L\,(Dh/g + c)$ where W = vehicle weight (lb), W_F = weight on front wheels (lb), L = wheelbase (in.), D = rate of deceleration of vehicle (ft per sec^2), c = distance from rear wheel to the center of gravity of the vehicle (in.), h = height of the center of gravity above the road surface (in.), g = gravitational conversion factor (32.2 ft per sec^2).

transferred W_t to the front wheels during braking times the length of the wheelbase L. A balanced system of forces, therefore, produces an equation $W_t L = Fh$, from which the transferred weight W_t can be obtained. The transferred weight always acts at right angles to the road surface, regardless of whether the vehicle is on a level or an inclined surface.

Since the braking effort is directly proportional to the weight on each wheel and the tire-to-road coefficient of friction, the proportion of braking done by the front wheels is considerably greater than for the rear wheels, especially at higher rates of deceleration. To obtain the maximum braking available, brake effort distribution should equal the dynamic weight distribution. Since, however, the weight distribution varies with the deceleration rate, this desired braking effort is not obtainable. Consequently, the full maximum braking effort is obtainable only under the one ideal set of conditions where the deceleration rate is such that the dynamic weight distribution is equal to the brake effort distribution.

Fig. 2 shows a graphical method of determining brake effort distribution. The deceleration rate chosen for maximum performance is 12.88 ft per sec^2, or when the tire-to-road coefficient of friction μ is 0.4 This gives a weight distribution of 60.1 per cent of the total vehicle weight on the front wheels. The brake, therefore, would be designed so that the front wheel braking effort would be approximately sixty per cent of the total braking effort. This can be accomplished best by proportioning the wheel cylinders on each wheel so that their respective force-applying areas are in proportion to the desired braking effort. (The wheel cylinder receives hydraulic brake fluid under pressure from the master cylinder when the brake pedal is depressed. Each wheel cylinder has two horizontally opposed pistons which, when moved outward by hydraulic pressure, make contact with the brake shoe assemblies and force them against the rotating brake drum's inner surface.)

Brake Size

After analyzing the deceleration requirements, the brake size required to insure adequate braking must be determined. Theoretically, the brake drum diameter should be large enough to dissipate the heat produced in severe braking without any pronounced change in a vehicle's braking ability. From the practical viewpoint, however, this is quite impossible with the internal-type brake because of space limitations and, in severe cases, overheating of the brakes can take place.

The diameter of the brake drum is definitely limited by the size of the wheel and the tire cross section. The drum width also has its limitations due to the closeness of other mechanisms, drum distortions, and weight limitations. In addition, the cooling possibilities are hampered by the recent trends to such styling features as lower chassis, deep fenders, and wide bumpers which tend to restrict the flow of cool air around the outside of the brake drum.

The ratio of the brake drum diameter to the wheel diameter is one of the major determining factors in obtaining a high mechanical advantage in the system. A larger drum diameter also results in more exposed drum surface area to dissipate the heat which is absorbed by the drums while braking. For these reasons, the brake drum diameter should be as large as possible with the determining factor being the amount of space available inside the rim of the wheel.

The brake drum width and, consequently, the brake lining area do not alter the effectiveness of the brakes because the coefficient of friction is independent of the area of contact between the brake lining and the brake

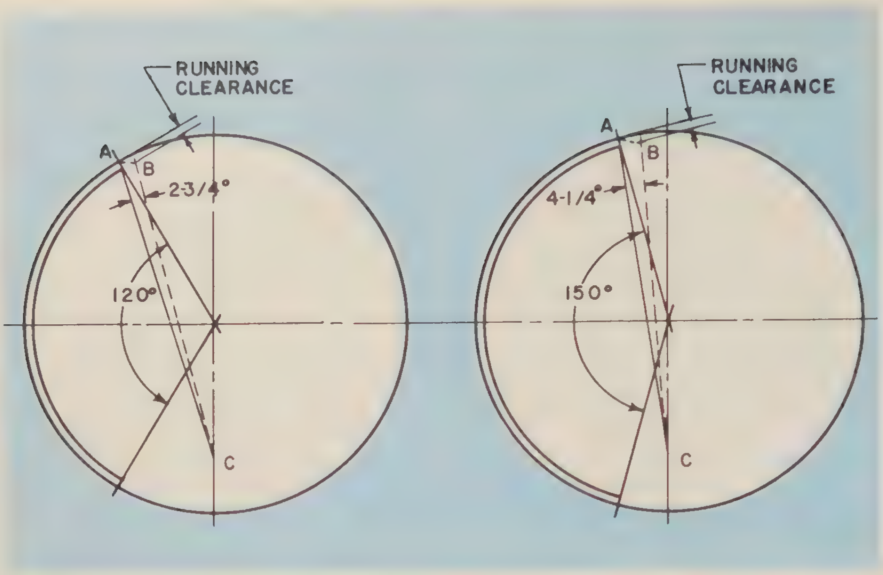


Fig. 3—The action of the brake lining of 120° arc length and 150° arc length is compared in this diagram. When the brake is in the running position, proper clearance must be maintained between the drum and the toe of the brake lining. For the same clearance at the toe of the lining, the 150° shoe must rotate about the anchor pin through an arc of 4¼° to make contact with the drum, whereas the 120° shoe requires an arc of only 2¾°. Since the angular travel of the shoes is proportional to the pedal travel, the 120° lining is the preferable design.

drum. The primary advantage of increasing the brake lining area is longer life and better heat dissipating capacity. The unit pressure imposed on the lining governs its life and an increase in the lining area decreases the unit pressure as the total pressure applied remains constant. With greater braking on the front wheels than on the rear wheels it is common practice to use wider linings on the front wheel brakes. This reduces the unit pressure on the front brakes to a value nearer to the unit pressure on the rear brakes.

Since the brake lining area governs the heat dissipating capacity of the brakes to a great extent, it should be proportional to the gross vehicle weight. Therefore, the vehicle weight to lining area ratio, or weight on each wheel divided by its effective brake lining area, is an important factor in determining lining life. A design having inadequate lining area will tend to have too great a temperature rise while braking, resulting in a lower coefficient of friction with a consequent reduction in brake effectiveness.

Brake Lining Arc of Contact

Brake lining area is the product of the lining width and the length of arc subtended by the lining. The length of arc, in turn, is dependent upon the brake drum radius and the angle of lining contact which is limited to approximately 120°.

There are several reasons why it is impractical to use over 120° of brake lining arc of contact. One reason is that when the toe of the brake shoe is too near the centerline of the applied force and the brake shoe anchor pin, the shoe becomes too highly self-actuated and tends to become "grabby" or self-locking. Self-actuation or self-energization is present to a certain extent in most brake designs and is due to the tendency of the forward acting brake shoe to rotate about the anchor pin and into the brake drum, thus increasing the total force acting on the shoe.

Fig. 3 illustrates another pertinent fact that justifies the limiting of the lining arc to 120°. The two brakes shown are of identical design, except for brake lining length. In order for the lining to obtain proper running clearance at the toe of the shoe, the 150°-arc-of-contact-lining must rotate about the anchor pin through an arc of 4¼° to come into contact with the drum. On the other hand, the 120° lining—with the same running clearance—swings through an angle of only 2¾° to make contact with the drum. This ratio of angular rotation is proportional to the pedal travel required to apply the brakes. It would require, therefore, 1.545 times as much pedal travel to take up the clearance in the 150° shoe as in the 120° shoe. Since the pedal travel and the mechanical advantage in the linkage are quite limited, the longer lining length is undesirable because it would necessitate sacrificing pedal travel to obtain slightly longer wearability.

Another reason for limiting the brake lining arc of contact to 120° is that the maximum pressure applied to the brake drum is at or near a point normal to the centerline through the anchor pin and the applying force. Fig. 4 illustrates a pressure distribution analysis of a simple brake shoe having a 120° arc of contact brake lining and shows that the radial pressure on the brake lining varies approximately as sin Θ. For example, a point **B** on the lining surface is chosen. When the brake is applied, this point moves along a line BD at right angles to the line point **B** makes with the brake shoe anchor pin. This motion can be broken down into a tangential pressure component CD and a radial pressure component BC. As pressure is applied, the lining touches the brake drum. Any further motion results in compression of the lining and expansion of the drum. This compression and expansion are proportional to the radial motion BC which takes place. The pressure, therefore, at any point, such as **B**, is directly proportional to the corresponding radial pressure BC at that point. The radial pressure, therefore, and also the local pressure vary as sin Θ. Since the sine has a value of unity at an angle of 90°, the maximum unit pressure is at a point normal to the brake shoe anchor pin and decreases as a function of sin Θ on both sides from this point. On a symmetrically located lining having a contact arc of 120°, the pressure at the ends of the lining is one-half as great as at the center. Because lining wear is a function of the radial pressure there is correspondingly less wear toward the outer ends of the shoe. In view of this, the added expense of a lining covering a larger arc of contact would hardly be justifiable.

Brake Lining Coefficient of Friction

Brake lining with a constant coefficient of friction under all operating conditions would simplify brake design problems greatly as this type of lining would yield a stable and uniform output throughout its useful life. At the present time, however, no suitable lining material having such frictional characteristics is available.

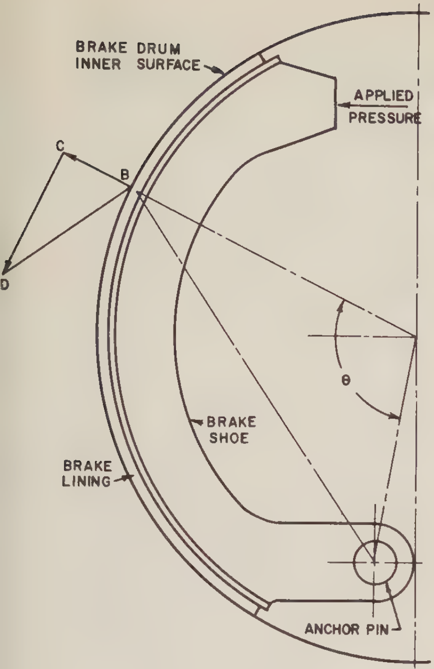


Fig. 4—On a symmetrically located brake lining having a 120° arc of contact with the brake drum, the pressure at the ends of the lining is one-half as great as at the center, resulting in less wear toward the outer ends of the brake shoe. This fact can be substantiated by choosing a point B on the brake lining. When the brake is applied, the point moves along a line *BD* at right angles to the line it makes with the brake shoe anchor pin. This motion can be resolved into a tangential pressure component *CD* and a radial pressure component *BC*. The compression of the lining and expansion of the brake drum, due to increasing pressure, is proportional to the radial motion *BC* which takes place. The pressure at point B is directly proportional to the corresponding radial pressure *BC*, which is a function of the sine of Θ. Since the sine is unity at an angle of 90°, the maximum unit pressure is at a point normal to the brake shoe anchor pin and decreases as a function of sin Θ on both sides from this point.

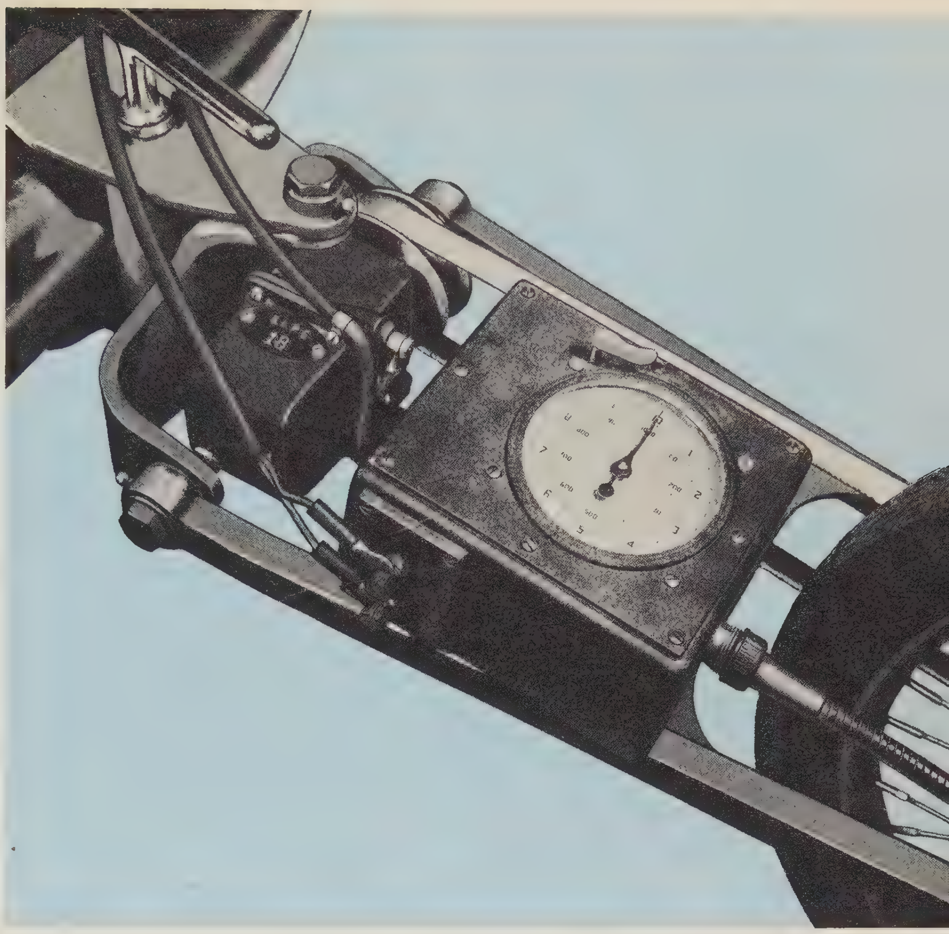


Fig. 5—A fifth wheel device is used at the General Motors Proving Ground in Milford, Michigan, to measure stopping distance. A specially constructed frame is mounted to a bicycle wheel which is towed by the test car. The wheel drives a solenoid-operated tachometer and odometer which accurately measure the time and distance required to stop.

The designer, therefore, must cope with linings having a variable coefficient. Brake linings having a higher than normal coefficient of friction will yield a greater torque output per pound of application force, but these linings also have a less stable output.

The lining-to-drum coefficient of friction varies with such factors as temperature, specific pressure, condition of the brake drum, and condition of the brake lining surface. Temperature changes have the greatest effect on changing the coefficient of friction under normal conditions because during braking application the temperature at the friction line may rise several hundred degrees. Another condition which seriously affects braking ability is the presence of water or grease on the lining's friction surface, in which case the coefficient may be lowered to a point where very little stopping ability is available.

As previously stated, the braking capacity of the front brakes should be greater than that of the rear brakes because of the weight transfer while stopping. A maximum brake drum diameter is desirable so it would not be advisable to reduce the drum diameter of the rear brakes to obtain increased braking capacity on the front brakes. Two alternatives are left to the designer: (a) use a lining with a higher coefficient of friction on the front brakes or (b) increase the wheel cylinder force applying area on the front wheels in proportion to the brake effort distribution desired. The use of linings with a higher coefficient of friction ordinarily is not recommended because they have greater self-actuating tendencies and, therefore, are less stable than linings having a low coefficient of friction. If linings having the same frictional characteristics are used on both the front and rear wheels, the proper proportioning of front wheel to rear wheel braking effort is best obtained by proportioning the wheel cylinder areas in relation to the desired brake distribution.

Brake Pedal Effort

The primary objective of the designer is to obtain a brake that will have certain desirable operational characteristics. One of the foremost among these is brake pedal effort which must be relatively light so as not to exceed the physical limitations of the driver. A light pedal pressure tends to result in brakes having less uniform response and less durability. Care must be taken, therefore, not to overemphasize the physical limitation aspect.

There are four available ways to

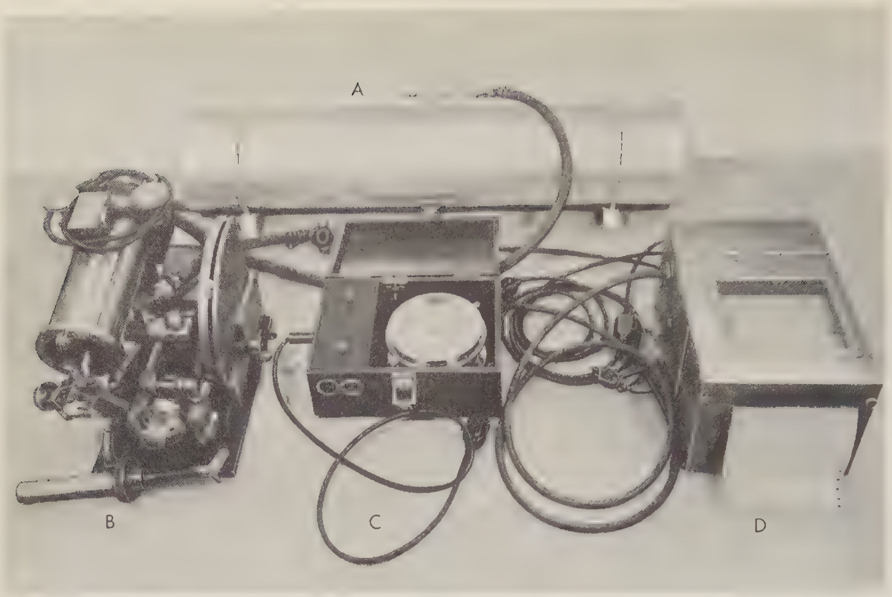


Fig. 6—To obtain information on brake performance characteristics under varying brake pedal pressures, a mechanical applicator or "mechanical foot" and a recording decelerometer are used. In operation, compressed air from a storage tank (*A*) is admitted to a cylinder (*B*). The cylinder actuates a piston which bears directly on the brake pedal. The operator is able to accurately control and adjust the amount of pedal pressure applied to bring the vehicle to a stop. The actual stopping distance is measured by a recording decelerometer unit (*C* and *D*). The decelerometer has a moving weight inside that tends to swing forward when the car begins to decelerate. The travel of this swinging weight is recorded on a moving paper tape to provide a permanent printed record. At the same time, the decelerometer also prints a record of time, pedal pressure, and pedal travel on the same tape. From these statistics test engineers determine the time required for a car to stop from any speed and with any amount of pressure applied to the brake pedal.

obtain a lighter pedal pressure without resorting to power mechanisms to actuate the brakes:

- Increase the mechanical advantage of brake drum diameter to wheel diameter
- Increase the mechanical advantage of the brake operating system
- Increase the efficiency of the brake operating system
- Increase the amount of self-actuation.

Little can be done to decrease pedal pressure from the standpoint of increasing the mechanical advantage of the brake drum diameter to the wheel diameter because the wheels in use are standardized to a great extent and the wheel rim contour and diameter limit the brake drum diameter obtainable.

The brake operating system presents the best means for increasing mechanical advantage. A large leverage ratio between the brake pedal and the brake shoe is very desirable because it gives decreased pedal pressure over the entire range of braking without affecting the stability and uniformity of performance. This method of increasing the output is limited by the total amount of pedal travel available, since some reserve must be left for lining wear unless some method of automatic self-adjustment could be devised. In this case, the reserve amount of pedal travel left for lining wear could be utilized to give a lighter pedal pressure.

Increasing the efficiency of the brake operating system presents another good method for obtaining lighter pedal pressure. Losses in the operating system include such things as friction in the mechanical linkage and resistance to flow of the hydraulic brake fluid. These losses are relatively small. They are, however, in the braking range where the reduction in pedal pressure is most desirous.

Reducing pedal pressure by employing more self-actuation is not regarded as a satisfactory solution in most instances. This is because the output gained through self-actuation is an exponential function of the coefficient of friction of the lining. As a result, any variation in the coefficient of friction would make a considerable change in the output and, in turn, make the brakes too sensitive for safe driving. The other methods for lowering pedal pressure, on the other hand, are all within the control of the driver because the output is directly proportional to the effort applied by the driver.

Power Booster Mechanisms

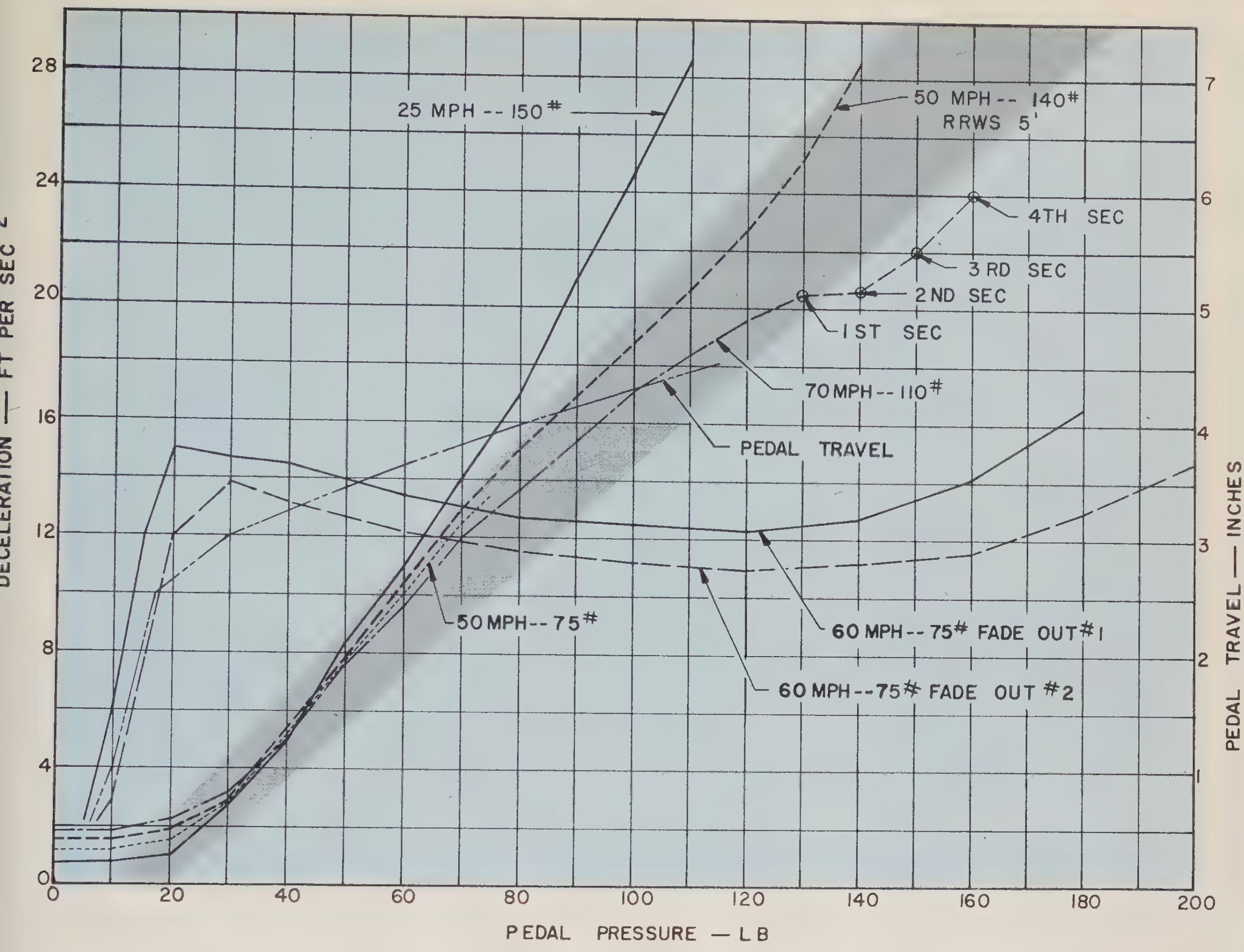
Since the aforementioned methods for reducing pedal pressure are all limited in the majority of modern automobile designs, the trend in recent years has been toward the utilization of power booster mechanisms to obtain lower pedal pressures without sacrificing uniformity of response in the braking system. In addition, the pedal travel can be reduced proportionately, thus giving a more desirable height relationship between the brake pedal and the accelerator pedal.

Another advantage of the power booster is that it is a self-contained unit that can be mounted directly to the toe pan or dash panel. As a result, all necessary linkage is on the interior of the body and is not exposed to road splash and dirt, as is the linkage in the conventional brake system. Frictional losses in the operating system, therefore, are kept to a minimum. Thus, as the trends in automotive design continue to restrict the other methods for obtaining the desired operational characteristics, the power booster comes into prominence as a desirable aid to the solution of the designer's problems.

Brake Testing

The testing phase in the development of a braking system for a new car model usually begins in the laboratory with test equipment set up to simulate service conditions. The principal laboratory brake-test equipment consists of a large diameter flywheel made of dynamically balanced steel plates rotated by an electric motor. A shaft on the opposite side of the flywheel is fitted with a mounting flange to which a brake drum can be secured. A brake backing plate, complete with brake shoes and actuating devices, is mounted on a pedestal to make a complete brake assembly.

In operation, the flywheel is brought up to speed by the electric motor. The brake is then applied to absorb the kinetic energy of the rotating mass of the flywheel. The flywheel is so constructed that its mass moment of inertia, and



ig. 7—Information obtained from the decelerometer recording unit is used to plot brake performance haracteristics curves which give an accurate indication of a specific brake design's ability to function nder varying deceleration rates and pedal pressures. The shaded area indicates good average brake erformance.

onsequently the kinetic energy it will ossess, can be varied to make it equiva- ent to that of a wheel on an automobile otating at various car speeds. This is ccomplished by increasing or decreasing he number of dynamically balanced steel lates which comprise the flywheel.

A testing machine of this type may be onnected to a control board through ydraulic and electronic controls similar o an engine dynamometer so that auto- natic cycling and recording can be btained. The testing machine can be ised to study the performance and char- cteristics of brake linings and drums, to letermine drum and lining wear, and to tudy heat flow and temperature gra- lients in brakes under various loading onditions.

Other laboratory tests on brakes are primarily cyclic in nature in which tests are conducted to determine the life of individual parts in the brake assembly.

Proving Ground Tests

After the brake design has passed the major development phase, tests are made with the brakes installed on a vehicle to evaluate them accurately before they are released for production. The major test in this category is to determine the available stopping ability.

Two methods for accurately measuring the stopping distance of a vehicle from a given speed are in common use today. One is the gun method in which a gun loaded with powdered chalk is fastened in a vertical position with the muzzle down to the side of a vehicle. When the pedal is depressed to apply the brakes, the gun is automatically fired and leaves a powder mark on the pavement at the point of initial brake application. The stopping distance required is measured from the powder mark to a point directly below the gun on the stopped vehicle.

Fig. 5 shows a fifth wheel device, developed by engineers at the General Motors Proving Ground, Milford, Michigan, which also is used to measure stopping distance. This device consists of a bicycle wheel, towed by the test car, which drives a solenoid-operated tachometer and odometer to measure accurately the time and distance required to stop.

Another instrument developed at the GM Proving Ground for studying brake performance characteristics is the recording decelerometer (Fig. 6). A mechanical applicator device, or "mechanical foot," is mounted between the brake pedal and the front seat. Compressed air from a storage tank is admitted through a governor-controlled, pressure-regulating valve to a cylinder. The cylinder, in turn, actuates a piston which bears directly on the brake pedal. This allows the operator to increase the pedal pressure at any desired constant rate. A paper-feed mechanism with a controlled speed of 5 in. per second, a deceleration recording unit, a pressure recording unit, and two solenoid markers for recording time and pedal travel comprise the recording unit. A complete record of brake performance characteristics for each stop—using either constant pedal pressure or varying pedal pressure from zero to a maximum of 200 lb—can be obtained with the pneumatic cylinder unit.

Fig. 7 shows sample curves plotted from data obtained when the recording decelerometer was used in testing passenger-car brakes. The "bogey," or shaded area, indicates good average brake performance. For example, brakes in which all of the curves (except the 25 mph-150 lb pressure curve) fall within the range of the bogey are considered to have good performance characteristics.

In making test runs for the data on the curves in Fig. 7, six deceleration stops were made. Sufficient time was allowed between stops for the brakes to cool to a normal operating temperature. The first run was made at 25 mph with the pedal pressure increasing to a maximum of 150 lb. The curve obtained in this run was above the bogey area. The second run was made at a speed of 50 mph with the pedal pressure varying up to 140 lb. Decelerations in this run simulated emergency braking and, in many instances, one or more of the wheels slid on the pavement just before the vehicle came to a complete stop. When this happened, the distance the wheel skidded was recorded and then indicated on the graph. Pedal travel data were also obtained on this run. Excessive pedal travel indicated that either the brake linings were worn and required adjustment or that there was excessive deflection in the brake linkage.

A third run was made at 70 mph with the pedal pressure changing at a constant rate up to a maximum of 110 lb. The pressure was held at this value until the car stopped. On the graph, the reading for each second of elapsed time from the 110-lb pedal pressure point until the time the vehicle stopped is indicated. A fourth run was made at a speed of 50 mph and with the pedal pressure increasing to 75 lb, which is considered typical of the braking encountered in normal driving. The curve on the graph for this run is plotted to the point where the pedal pressure reached the preset maximum of 75 lb.

In recent tests, a 90-mph run has been added to show high-speed braking and the 50 mph-75 lb pressure run eliminated. The 90-mph run is similar to the 70-mph run except that the pedal pressure changes at a constant rate up to 90 lb. The pressure is then held at this value until the car stops.

Two runs are also made in succession at 60 mph and a pedal pressure of 75 lb applied immediately and held constant until the car stops. No time is allowed for the brakes to cool between the two runs. These runs give an indication of the fade characteristics of the brakes. A brake lining showing considerable tendencies to fade would be unsatisfactory because of its sensitivity to temperature changes. Test results obtained from an instrument such as the decelerometer are very valuable to the engineer in enabling him to compare several proposed lining combinations and select the one which shows the most desirable characteristics.

A 12-stop fade test also is run to obtain information on fade and recovery characteristics of a specific brake. In this test, 12 successive stops, with the transmission in neutral, are first made for fade characteristics from 60 mph at a deceleration rate of 15 ft per sec^2. The initial and final line pressures are recorded for each stop. Total elapsed time for the stops and any remarks such as pulling, noise, and odor also are recorded. Immediately following the fade runs, brake recovery is determined by making slow-down checks from 40 mph, at a deceleration rate of 8 ft per sec^2, at 1-mile intervals for at least 12 stops or until the line pressure stabilizes. Line pressures and any pertinent remarks also are recorded on this test.

Another important step in testing brakes is road trip evaluation. Test cars equipped with the proposed brake installation are taken on road trips through rugged mountainous terrain where the brakes must be snubbed as many as 20 or 30 times to negotiate curves in descending the mountain grades. In snubbing the brakes, the car is decelerated at rates of approximately 15 ft per sec^2 from speeds of 60 mph down to speeds ranging from 40 mph to 25 mph to negotiate the curves. A test of this type gives the engineer a reliable indication of the fade and recovery characteristics of the brakes under conditions that approach the maximum to which a brake would be subjected, even in the most rugged driving.

Other tests to which brakes may be subjected include endurance tests for lining wear, burning of the linings, scoring of the drums, and dust sealing. These tests are often run in conjunction with other tests on the vehicle, thereby yielding results that are reasonably representative of those encountered under normal driving conditions.

Summary

In the design of an automotive brake system the engineer must take many factors into consideration, often compromising the theoretical with the practical aspects of the design to obtain the maximum available braking performance. By a careful evaluation of these factors in the design phase, the engineer can recommend a proposed brake design with a reasonable assurance that it will perform satisfactorily under actual operating conditions. This anticipated performance then must be verified experimentally in the laboratory and later at the proving ground to assure that maximum braking has been obtained and that the braking response is as stable and as uniform as possible under all operating conditions.

Bibliography

Other related literature in this field includes the following:

BACHMAN, B. B., "Analysis of Brake Design Factors Gives Basis for Future Studies," *S.A.E. Journal*, Vol. 33, No. 3 (September 1933), pp. 298-304.

BURKHARDT, O. M., "Fundamentals of Brake Design," *S.A.E. Journal*, Vol. 17, No. 1 (July 1925), pp. 64-73.

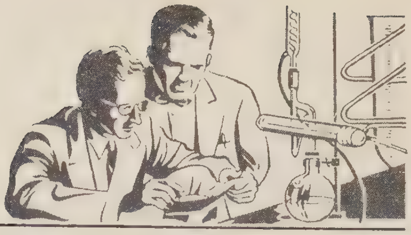
CHASE, T. P., "Passenger Car Brake Performance—Limitations and Future Requirements," *S.A.E. Quarterly Transactions* (January 1949), pp. 26-40.

FREHSE, A. W., "Fundamentals of Brake Design," *S.A.E. Journal*, Vol. 27, No. 2 (August 1930), pp. 170-78.

HOUSE, B. E. and GOEPFRICH, R. A., "Brake Design Trends," *S.A.E. Journal*, Vol. 52, No. 9 (September 1944), pp. 436-44.

Notes About Inventions and Inventors

By **EDWIN S. DYBVIG**
Patent Section
Dayton Office Staff



Growth of the patent system parallels that of industrial progress

THE patent system was evolved from the need of some inducement to those who made a discovery to disclose it to the public so that the discovery and its benefits would not die with the death of the party making the discovery. While it is true that some of the important discoveries were handed down from father to son as a family secret and thus were perpetuated, handing down in this manner was not conducive to widespread dissemination of the information since it was necessary for the preservation of the family secret to limit the secret to a very small number. It is no wonder that under such a practice very little real industrial progress was made.

Early Patent Grants

The practice of rewarding inventors for important contributions was perhaps started before the time of recorded history and was sporadically practiced in various forms and with varying degrees of success from time to time down through the ages. It is reported, for example, that the Greek city of Syboris which was destroyed 500 years before the Christian Era had a law which provided that "if any confectioner or cook invented a peculiar or excellent dish, no other artist was allowed to make it for a year." Rome rewarded some of its citizens for outstanding contributions—including inventions—by granting permission to such individuals to levy taxes upon some conquered and subject nation; however, all too frequently these rewards were given on the basis of personal friendship of an emperor, rather than for meritorious contribution. Hard-pressed monarchs in medieval Europe could not afford to give up any of their rights to tax subjects and, hence, began a system of granting monopolies which were secured to the recipient by a proclamation which became known as a patent. Henry III possessed the right to issue such patents and exercised the right sparingly and without any great abuse but, by the time of Queen Elizabeth I, the right became unpopular because of its abuse by the promiscuous granting of monopolies on such articles of common necessity as salt, iron, and a multitude of other well known commodities in everyday use. The fact that these monopolies were granted in great numbers to court favorites did not add to their popularity. During the reign of James I, all monopolies were made illegal by the "Statute of Monopolies," except as applied to new manufactures or inventions. This, then, formed a sound basis for our modern patent system.

1,000 Years Without Change

Prior to the formation of a sound patent system, progress was indeed slow. From the year 400 A.D. to the year 1400 A.D., no appreciable industrial improvements of any kind were made. Thus, the mode of transportation, communication, home construction, and so forth remained substantially the same for 1,000 years. This lack of progress cannot be ascribed to a scarcity of potential inventors or a scarcity of natural resources as both existed side by side. In this land of ours, with all of its resources, the American Indian hadn't even invented the wheel.

The founders of this country had the foresight to provide for a patent system. Even before the Constitution of the United States was adopted, many of the American colonies and states granted patents. The first patent on this continent was granted by the Massachusetts General Court to Samuel Winslow in 1641. On April 10, 1790, George Washington signed the bill which laid the foundations of the modern American patent system. Under this patent system, progress has been made at an accelerated rate with the result that, within the last 100 years, we have witnessed the introduction of electricity, telegraphy, telephony, radio, television, automobiles, airplanes, cyclotrons, and atomic- and hydrogen-powered devices, as well as many other processes and products too numerous to mention.

Looking Toward Tomorrow

It would be preposterous even to attempt to predict the inventions which will be made in the future. We do know that some of the less practical inventions of today will be developed to the practical stage very quickly. Basic inventions frequently have been made long before the world has been ready to derive any benefits from them. The electric motor, for example, was invented at a time when the only source of electricity for operating the motor was the galvanic cell or chemical battery. This led at least one authority in that day to predict that the electric motor would never be used extensively and could not replace the steam engine for general use, because the amount of power available in a dollar's worth of zinc was only a fraction of that which was available in a dollar's worth of coal. The rest of the story of the electric motor needs no telling.

From the many lessons of the past we can be reasonably certain that so long as our type of patent system is allowed to function properly it will continue to promote inventions and the widespread dissemination of the latest and most advanced information concerning these inventions for the good of all mankind.

On this and on the following pages are listed some of the patents granted to General Motors prior to April 30, 1955. The brief patent descriptions are informative only and are not intended to

***Inventors' names marked with an asterisk in this section have had their biographies published in a previous issue of Volume 2, GENERAL MOTORS ENGINEERING JOURNAL.**

define the coverage which is determined by the claims of each patent.

Patents Granted

- **Kendall O. Bower**, *Electro-Motive Division, LaGrange, Illinois, for an Overtravel Compensating Link with Safety Spring, No. 2,702,331, issued February 15.* This invention relates to a compressive spring link between driving and driven members having its ends slidably guided in a groove in one of the members to maintain the spring operative if it breaks.

Mr. Bower serves as a-c control engineer in Electro-Motive's Engineering Department. He was first employed in that Department in 1945 as a project engineer. He was promoted to assistant control development engineer in 1951 and to his present duties in 1953. Mr. Bower's current assignments cover the development of control equipment for mobile electric automatic generating equipment. Iowa State College granted him the B.S. degree in electrical engineering in 1938. Mr. Bower is a member of the American Institute of Electrical Engineers.

- **Ellwood F. Riesing**, *Inland Manufacturing Division, Dayton, Ohio, for a Pressure Indicating Device, No. 2,704,045, issued March 15.* This invention relates to a transparent plastic valve cap for a tire. The cap has an externally visible indicator therein which is movable and responsive to indicate the pressure within the tire.

Mr. Riesing is assistant chief engineer at Inland Manufacturing. Formerly manager of Automotive Engineering and Sales, Firestone Industrial Products Division of Firestone Tire and Rubber Company, Mr. Riesing was employed by the Division in 1952 on special assignments for the general manager. Two years later he was appointed to his present position. He has written numerous papers on the applications of rubber in the automotive industry, and is an author for the most recent Kent's Mechanical Engineer's Handbook. Currently, he is a member of the S.A.E., A.S.M.E., and the Engineering Society of Detroit.

- **Dr. Edward J. Martin* and Walter E. Sargeant***, *Research Laboratories Division, GM Technical Center, Detroit, Michigan, for a Cable Insulation Tester, No. 2,704,825, issued March 22.* This patent relates to a cable insulation tester which features a novel test head construction through which the cable is drawn during the test for application of a high-frequency test voltage to the insulation thereof.

- **Dr. Howard W. Pearsall**, *Research Laboratories Division, GM Technical Center, Detroit, Michigan, for a Process for Removing Deposits from Internal Combustion Engines and Composition Therefor, No. 2,704,733, issued March 22.* This patent pertains to a new liquid composition and process for removing adherent carbonaceous deposits from metal surfaces, such as walls of gasoline engine combustion chambers.

Dr. Pearsall is a research chemist in the Fuels and Lubricants Department of the Research Laboratories, a post he has occupied since joining the Division in 1949. Dr. Pearsall has been engaged in chemical research for more than twenty years, including his former associations with the Ethyl Corporation Research Laboratories and Wyandotte Chemicals Corporation. Some of this work has related to air analyses, exhaust-valve burning, hydrocarbon synthesis, and tetraethyl-lead synthesis. He received the B.S.Ch.E. degree from University of Missouri (1931) and the Ph.D. degree from Wayne University (1948).

- **Clair A. Short, Jr.* and Grover N. Conley**, *Aeroproducts Operations of Allison Division, Dayton, Ohio, and Allison Division, Indianapolis, Indiana, respectively, for a Jet Nozzle Actuator, No. 2,704,436, issued March 22.* This patent relates to a jet nozzle actuator mechanism which features a self-adjusting crosshead that prevents binding of the parts of the actuator mechanism.

Mr. Conley is assistant supervisor in Allison's Quality Control Department. He has been with the Division since he was first employed in 1948 as a senior project engineer in its Engineering Design Department. He transferred to the Purchasing Department three years later and reached his present position in 1953. As such, he is responsible for quality control of material purchased for Allison's Aircraft Engines Operations. Mr. Conley earned the B.S. degree in mechanical engineering from University of Kentucky in 1939 and he is a registered professional engineer, State of Ohio.

These patent descriptions are informative only and are not intended to define the coverage which is determined by the claims of each one.

- **William J. deBeaubien***, *Pontiac Motor Division, Pontiac, Michigan, for an Optical Viewing Device, No. 2,704,962, issued March 29.* This invention relates to an optical viewer for automobiles which is formed by a transparent prism having surfaces so positioned that the light rays are incident on the lower surface at an angle greater than the critical angle of the material of which the body of the prism is formed, thereby reflecting the light rays.

- **Arthur W. Hills**, *AC Spark Plug Division, Flint, Michigan, for Illuminatable Signals, No. 2,705,317, issued March 29.* This invention relates to means for uniformly illuminating an instrument having transparent indicia such as are employed on an automotive instrument panel. In order to obtain a uniform illumination of the indicating area free from any glare, the source of illumination and the indicia are disposed on the opposite sides of a prism so the light passes through a prism having one side thereof peculiarly shaped to disperse the light throughout the prism so that the light from the prism will be directed uniformly over the entire area of the indicia.

Mr. Hills serves as senior designer at AC Spark Plug. His 36 years of design experience in General Motors date from his employment with Buick Motor Division, Flint, Michigan, in 1919. During that period, his work centered on axle assembly. In 1927 he transferred to AC Spark Plug where he progressed from artist and designer to his present position. For the past 28 years Mr. Hills has devoted his major attention to designing instrument clusters. His work in this area has resulted in four patents covering instrument cluster lighting.

- **Peter J. Jorgensen and Clarence H. Jorgensen*** *AC Spark Plug Division, Milwaukee, Wisconsin, for a Mechanism for Controlling the Starting and Operation of Internal Combustion Engines, No. 2,705,484, issued April 5.* This patent is directed to the automatic choke and covers an unbalanced choke valve subject to the pressure differential across the valve created by flow of air into the carburetor, which is also operated by a thermostat responsive to variations in engine temperature and a suction-operated member connected to the valve through the thermostat and operable in response to variations in engine suction posterior to the throttle.

This particular patent covers the basic automatic choke, still in use on present-day cars and developed by the late **Peter J. Jorgensen** and his son, **Clarence H. Jorgensen,** at present administrative engineer at AC Spark Plug. An exclusive royalty agreement was entered into in 1932 with Delco-Remy Division and the Bendix Aviation Corporation. This device was put into production by Delco-Remy in 1933. General Motors acquired this patent by purchase in 1937 from Clarence H. Jorgensen, acting on behalf of the estate of Peter J. Jorgensen and on his own behalf. This patent is unique in that it was over twenty-three years from the filing date (1932) until the patent issued (1955). (The average span between filing and issuance is about three and one-half years; some patents require less than one year.) In 1914 Clarence H. Jorgensen became associated with his father, and they operated their own manufacturing and engineering organizations until the senior Mr. Jorgensen's death in 1934. During this period several mechanical innovations were pioneered which still are in use—water temperature controls for automatic washers, priming device for ordnance vehicles, the automatic choke, a manifold heat control, and a thermostat for reguating engine water temperature.

- **Frederick W. Sampson and Allen L. Everitt,** *Inland Manufacturing Division, Dayton, Ohio, for a Shock Absorber, No. 2,705,634, issued April 5.* This patent relates to a double-acting shock absorber comprising a cylinder having a piston disposed therein and impact responsive means for frictionally opposing relative movement therebetween.

- **Frederick W. Sampson,** *Inland Manufacturing Division, Dayton, Ohio, for a Method of Making a Steering Wheel, No. 2,705,816, issued April 12.* This patent relates to a method of making a reinforced steering wheel rim having two different molded materials, each constituting a discreet portion of the rim periphery.

Mr. Sampson is chief engineer at Inland Manufacturing. He joined the Division as assistant chief engineer in 1929, and was appointed chief engineer

in 1942. More than forty patents have been issued as a result of his work in the fields of automotive parts, ice cube freezing trays, and firearms. He is a 1924 mechanical engineering graduate of Cornell University and is currently affiliated with the Society of Automotive Engineers.

Mr. Everitt is a project engineer in the Engineering Department at Inland Manufacturing. In 1931 Purdue University granted him the B.S.M.E. degree and his employment with Inland Manufacturing began shortly afterward. His experience with this Division has covered drafting, estimating, laboratory, and design activities. His past projects have been on vibration control and he is currently engaged in firearms design involving experimental automatic weapons. Mr. Everitt is a member of an S.A.E. group on automotive rubber standards and the GM Automotive Standards Committee.

- **Bradley F. Watson,** *Frigidaire Division, Dayton, Ohio, for a Heat Exchange Unit, No. 2,705,616, issued April 5.* This patent relates to a removable air-cooled moisture condenser of sheet metal construction for use in removing moisture from the air discharged by a clothes dryer.

Mr. Watson is supervisor of the Tool Design and Tool Engineering Department of Frigidaire. In 1923 he joined Delco-Light Company as an apprentice in the Tool Design Department. When part of Delco-Light was absorbed by Frigidaire, Mr. Watson remained in the tool design field. He was promoted to foreman in 1939 and to general foreman in 1945. In 1951 Mr. Watson was made assistant supervisor of the Tool Design and Tool Engineering Department and was promoted to supervisor in 1953. He holds a certificate in tool engineering from the International Correspondence School.

- **Adolph F. Braun,** *Buick Motor Division, Flint, Michigan, for a Compound Carburetion System, No. 2,705,942, issued April 12.* This invention relates to a throttle-operating mechanism for a compound carburetor, the secondary throttle of which is operated by the primary throttle and by the suction created by air flow in the primary passage.

Mr. Braun is staff engineer, Experimental Engineering, in Buick Motor's Engineering Department. He received the B.S.M.E. degree from University of South Dakota in 1928 and in July of the same year was employed by Buick Motor as a student engineer. He advanced later to the positions of engineer, senior project engineer, and section engineer. In 1954 Mr. Braun was promoted to staff engineer-in-charge of the Buick Proving Grounds and all dynamometer and physical test work, as well as the Technical Data and Road Car Departments. This is the fourth patent resulting from Mr. Braun's research on carburetion systems. He is a member of Pi Sigma Pi, the Society of Automotive Engineers, and the American Ordnance Association.

- **Charles D. Holton,** *Buick Motor Division, Flint, Michigan, for an Auxiliary Vacuum Pump for Power Brakes, No. 2,705,870, issued April 12.* This patent relates to a control system for the vacuum-actuated brake booster wherein an independent vacuum pump is started to supply a vacuum source to the brake booster whenever the engine stalls or operates below a predetermined idle speed.

Mr. Holton is an engineer in Buick Motor's Engineering Department. He started with GM in 1933 as a test engineer at the General Motors Proving Ground, Milford, Michigan. Two years later he transferred to Buick Motor's Technical Data Department. His promotion to junior engineer the following year was succeeded by promotions to project engineer and brake project engineer. Mr. Holton's current assignment centers on future brake development. Past projects involved development on tank components, and automotive axles, clutches, and brakes. Mr. Holton is a 1931 mechanical engineering graduate of University of Michigan.

- **George C. Pearce,** *Frigidaire Division, Dayton, Ohio, for a Frameless Range Construction, No. 2,705,947, issued April 12.* This patent relates to a cooking range wherein the oven liner provides the nucleus for the mounting thereto of a plurality of component range parts, including the outer range side panels and the range top.

- **George C. Pearce,** *Frigidaire Division, Dayton, Ohio, for an Electric Range, No. 2,707,225, issued April 26.* This patent relates to a range oven which is provided with two doors which are pivotally mounted upon opposite sides of the oven for horizontal swinging movement. Special actuator means causes both doors to open simultaneously upon actuation of one door.

Mr. Pearce joined Frigidaire's Appliance Engineering Department 25 years ago. As a section head in the Appliance Engineering Department—his present position—he is supervisor of range and water heater design. Mr. Pearce's work in the fields of electrical controls and circuits has led to more than sixty granted patents and a published paper on the evaluation of performance for appliance controls. He received the B.S. degree in mechanical engineering from Stanford University (1924). In addition, he is a member of the Dayton Engineers' Club.

- **Joseph R. Pichler*,** *Frigidaire Division, Dayton, Ohio, for a Multiple Deck Refrigerated Display Case, No. 2,705,875, issued April 12.* This invention relates to an open multiple deck bin-like display case wherein food products in an upper bin are cooled to substantially the same temperature as foods stored in a lower bin by circulating refrigerated air forwardly over the upper bin and then rearwardly directly beneath this upper bin prior to its entrance into the space above the lower bin.

- **Garthwood R. Taylor,** *Aeroproducts Operations of Allison Division, Dayton, Ohio, for a Propeller Blade Mounting and Servo Actuator, No. 2,706,007, issued April 12.* This patent relates to a servomotor for varying the pitch of a propeller blade, wherein the propeller blade is rotatably supported on the outer periphery of a hub spindle and the servomotor is disposed with the hub spindle.

Mr. Taylor serves as a section head in Aeroproducts' Engineering Department.

He started with Allison in 1941 as a detailer, and promotions to junior engineer, assistant project engineer, project engineer, and senior engineer quickly followed. In 1951 he was made section head of Engineering Standards, supervising engineering facilities and test equipment, design and engineering quality control, and engineering standards. His previous work on hydraulic aircraft propellers has resulted in four granted patents. The Detroit Institute of Technology granted Mr. Taylor the B.S. degree in mechanical engineering in 1941.

- **Charles H. Beare, Stanley R. Carson, and Russel L. Monbeck,** *Inland Manufacturing Division, Dayton, Ohio, for a Method of Making Wire Reinforced Flexible Hose, No. 2,707,017, issued April 26.* This invention relates to a method for forming flexible tubing wherein a wire helix is fed directly to an extrusion head and wherein tape material is extruded directly around and over the wire to form a reinforced tape which is concomitantly wrapped around a mandrel in overlapping fashion to form a tube.

Mr. Carson is supervisor of experimental engineering at Inland Manufacturing. In January 1955 Mr. Carson marked his 34th year of service with General Motors. He was first employed in 1921 as a tool maker in the Tool Department at Inland Manufacturing. Later promotions to laboratory assignments led to his position as assistant superintendent and, eventually, to supervisor in the Experimental Engineering Department. Mr. Carson's work in the field of development engineering has resulted in 10 granted patents.

These patent descriptions are informative only and are not intended to define the coverage which is determined by the claims of each one.

Mr. Monbeck is an experimental extruding die maker in the Experimental Engineering Department at Inland Manufacturing. He was originally employed by the Tool Department of that Division in 1921, as a tool maker. In 1940 he was transferred to the Control Laboratory. His succeeding assignment was in the Experimental Engineering Department where he was promoted to his present duties in 1950. This is the first patent issued as a result of Mr. Monbeck's work in tool and die making.

Mr. Beare is no longer with the Division.

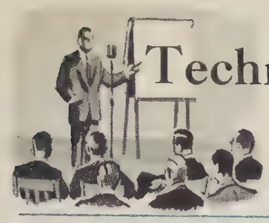
- **Charles H. Frick* and Ralph J. Rays,** *Detroit Diesel Engine Division, Detroit, Michigan, for an Engine Speed Governing Device, No. 2,706,975, issued April 26.* This relates to an improved linkage between an engine pneumatic governor and the fuel control member for better engine speed stability over the operating range.

Mr. Rays is no longer with the Diviion.

- **John H. Little and Harold J. Schoelles,** *Chevrolet Motor Division, Detroit, Michigan, and Harrison Radiator Division, Lockport, New York, respectively, for a Heater and Defroster Control Mechanism, No. 2,707,079, issued April 26.* The above patent relates to a heater and defroster control mechanism for automotive vehicles consisting of a series of control levers for controlling the air valves, the blower motor, and thermostat, the levers being so arranged that movement of the lever controlling the flow of air for defrosting purposes compels actuation of the control lever which operates the air inlet valve and the blower motor.

Mr. Little is assistant staff engineer in Chevrolet Motor's Engineering Department where his work is concerned primarily with advanced design of electrical components and accessories. He joined the GM Research Laboratories Division in 1925 as experimental engineer in the electrical engineering section. He remained with the Laboratories until 1933 when he transferred to Chevrolet Motor as assistant electrical engineer. After several advancements, he was appointed to his present post in 1952. He was granted the B.S. degree in electrochemical engineering by Massachusetts Institute of Technology in 1923.

Mr. Schoelles is no longer with the Division.



Technical Presentations by GM Engineers

The speaking appearance is only one of the channels through which technical information from General Motors laboratories and factories reaches the nation's engineering educators and students. GM engineers lecture in college classrooms and before civic groups, present papers, and serve as panel members at technical society meetings. These appearances are encouraged by the GM Divisions and Staffs as part of their program of sharing engineering developments with the public.

Air Conditioning and Refrigeration

Frederick L. Chase, superintendent of the Manufacturing Research Laboratories of Frigidaire Division, presented the talk "Chemical Magic"—story of the Freon refrigerants—on three separate occasions during the period covered in this report. On April 13 he spoke before the Tri Chapter (Cincinnati-Columbus-Dayton) meeting of the American Society for Metals; on May 24 he appeared before the Dayton Section of the American Society of Lubrication Engineers; and on June 21 he addressed the Fellowship Club in Dayton.

Atomic Energy

Dr. Helen Barlett, supervisor of ceramic research in AC Spark Plug Division's Engineering Department, spoke on "The Uses of Atomic Energy" before the Zonta Club in Flint, Michigan, on June 8.

On June 16, before the Exchange Club meeting in Flint, **Dr. James A. Norton**, senior experimental chemist in the Research Department of AC Spark Plug, talked on "Atomic Radiation—Curse or Blessing?"

Automotive Engineering

On June 21, at a meeting of the Institute of Traffic Engineers, **Howard K. Gandelot**, engineer-in-charge of the General Motors Engineering Staff's Vehicle Safety Section, talked about "Horsepower and Motor Vehicle Safety."

George Aitken, assistant staff engineer-in-charge of the Drafting Group at Chevrolet Motor Division's Central Office, addressed the New York University Student Section of the S.A.E. on May 16. Mr. Aitken talked on "The Corvette Plastic Body."

The May 17 meeting of the Technical Committee of the National Paint, Varnish and Lacquer Association in Atlantic City was the occasion for a talk by **Wardley D. McMaster**, assistant head of the Research Laboratories Division's Chemistry Department. Mr. McMaster gave "A Fancy Formula for Finer Films."

Kenneth E. Coppock, assistant chief engineer of Fisher Body Division's Engineering Section, addressed the Canadian Section of the Society of Automotive Engineers on May 27 in Oshawa, Ontario. The title of Mr. Coppock's talk was "Who is the Designer—the Stylist or the Engineer?"

On June 6, at a meeting of the Chicago Chapter of the Sheet and Tin Section, American Iron and Steel Institute, **Homer C. Pratt**, senior project engineer in Fisher Body's Engineering Laboratory, spoke on "The Selection and Processing of Sheet Steel for Auto Body Panels."

Dr. Robert F. Thomson, head of the Research Laboratories' Metallurgy Department, addressed the Metals Division of the Special Libraries Association meeting in Detroit on June 14. Dr. Thomson's subjects were "New Horizons in Metals and Their Use" and "Trends in New Material Development."

Several General Motors personnel participated in the S.A.E. Golden Anniversary Summer Meeting held in Atlantic City, New Jersey, from June 12 through 17. **Carl C. Jakust**, assistant staff engineer-in-charge of the Passenger Car Body Section of Chevrolet Motor, was a member of a round-table discussion on major body panels. **G. F. Roddewig**, experimental engineer in GMC Truck & Coach Division's Experimental Department, talked on "Cooling Systems." **H. A. Reynolds**, engineering supervisor, Radiator Section, Harrison Radiator Division, was a participant in a round-table discussion on automotive cooling systems. Three Research Laboratories' personnel—**James C. Holzwarth**, senior research engineer, Metallurgy Department; **Alfred F. Boegehold**, assistant to the general manager; and **Dr. Robert F. Thomson**, head of the Metallurgy Department—collaborated on the paper "A Study of Non-protective Rust Formation on Auto Body Sheet Steel."

Buses and Off-the-Road Vehicles

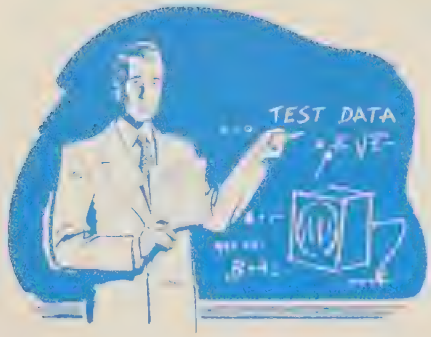
Alan S. McClimon, manager of sales development at Euclid Division, spoke at the Coal Convention of the American Mining Congress, held in Cleveland on May 17. Mr. McClimon discussed "Off-Highway Coal Haulers—Present and Future Developments."

Messrs. Nebel, Rounds, and Elliott of the Research Laboratories collaborated on the paper "The Composition of Exhaust Gases from Diesel, Gasoline, and Propane-powered Motor Coaches" presented before the Air Pollution Control Association's Annual Meeting in Detroit on May 25. **George J. Nebel** is a research engineer in the Fuels and Lubricants Department, **Fred G. Rounds, Jr.** is a senior research engineer in the same Department, and **Martin A. Elliott** is a Research Laboratories consultant.

Engines and Transmissions

J. A. Winter, supervisor of the Technical Data Section, Transmissions Engineering Operations of Allison Division, appeared at the May 15 meeting of the American Society of Tool Engineers in Richmond, Indiana. Mr. Winter talked on "The Introduction of Hydraulic Torque Converters."

"The Development of a Large, Two-Cycle Gas Engine" was the title of the talk given by **Eric R. Brater** before the Oil and Gas Power Division of the American Society of Mechanical Engineers meeting in Washington, D. C., on June 6. Mr. Brater is assistant chief engineer of Cleveland Diesel Engine Division.



At the 4th World Petroleum Congress held in Rome, Italy, **J. M. Campbell**, technical director of the Research Laboratories, on June 10, presented the paper "What's Happening to Automotive Engines." This paper was prepared jointly by members of the Research Laboratories' Automotive Engines Department—**Darl F. Caris**, head; **Archie D. McDuffie** and **B. John Mitchell**, assistant department heads; and **Floyd A. Wyczalek**, supervisor.

June 13, Messrs. Caris, McDuffie, Mitchell, and Wyczalek played an active part in the S.A.E. Golden Anniversary Summer Meeting by jointly submitting the paper "Mechanical Octane Numbers for Higher Efficiency."

Several other General Motors Divisional personnel presented papers at the S.A.E. Golden Anniversary Summer Meeting in the general areas of automotive engines and transmissions. **Rudolph J. Gorsky**, staff engineer—transmissions in Buick Motor Division's Engineering Department, talked on the "Buick's Variable Pitch Dynaflow." **R. E. Kaufman**, truck transmission engineer in Chevrolet Motor's Truck Chassis Group, was a member of a panel on power train compatability. **C. V. Crockett**, GMC Truck & Coach's chief engineer was a member of this same panel. **L. T. Flynn**, truck engineer, and **F. W. Brede, Jr.**, project engineer—both of GMC Truck & Coach's Trucks Department, jointly presented a description of the "General Motors Twin Hydra-Matic."

Foundry and Castings

On May 23, as part of the American Foundrymens Society convention held in Houston, Texas, **L. J. Pedicini**, project engineer in the Foundry of the Process Development Section, General Motors Manufacturing Staff, described "General Motors Experimental Foundries."

During the S.A.E. Golden Anniversary meeting—from June 12 through 17—**John G. Coffin** was a member of a panel on the physical properties of molded parts. Mr. Coffin is assistant metallurgical engineer in the Production Engineering Group of Chevrolet Motor.

General Engineering

Charles E. Fausel discussed "The Engineer's Role in Management" at the April 5 meeting of the Mineral Industry Society held at University of Illinois. Mr. Fausel is superintendent of the Grey Iron Foundry at Central Foundry Division's Danville, Illinois, plant.

On May 18 **M. O. Alsager**, supervisor of budgets in AC Spark Plug's Factory Accounting Department, described "Cost Accounting at AC Spark Plug" to the cost accounting class at Flint Junior College.

Richard J. Sullivan, research engineer in the Special Problems Department of the Research Laboratories, addressed the Industrial Mathematics Society in Detroit on May 18. The title of Mr. Sullivan's talk was "A Procedure for Obtaining Periodic Numerical Solutions to Simultaneous Non-linear Differential Equations."

Before The Ohio State University Student Section of the S.A.E. meeting on May 19, **John Burnell**, design engineer in the Engine Design Group of Chevrolet Motor, outlined "The Design and Development of an Engineer."

Earl Brohl, senior project engineer in AC Spark Plug's Automotive Engineering Department, talked on "Engineering Publications" at the May 23 meeting of the American Institute of Electrical Engineers Executive Committee, held in Chicago, Illinois.

As part of the Pittsford, New York, Central School Vocations Day program held on May 26, **Fred W. Cummings**, administrative engineer in the Product Engineering Department of Rochester Products Division, discussed "Engineering as a Vocation."

Requests may be directed to the Educational Relations Section for assistance in obtaining the services of GM engineers to speak before engineering classes or other student groups.

Leonard E. A. Batz, senior project engineer in the Defense Engineering Department of AC Spark Plug, addressed the Lumber Dealers Association in Flint, Michigan, on June 1. The title of his talk was "Engineering and You."

On June 23 **Kenneth A. Meade**, director of the Educational Relations Section, Central Office Department of Public Relations, presented the talk "Professional Engineering Ethics in Practice" at the American Society for Engineering Education Annual Meeting held at Pennsylvania State University.

Lubrication and Bearings

Before a group of aircraft engineers attending a symposium sponsored by New Departure Division at Burbank, California, on June 7, two of the Division's personnel presented papers. **Heinz Hanau**, project engineer in the Product Engineering Department, and **Martin D. Hecht**, project engineer in the same Department, collaborated on the paper "High Speed, High Temperature Bearing Development." On June 8 Messrs. Hanau and Hecht presented this same talk before an engineering group from North American Aviation, Incorporated meeting in Downey, California.

Bearings were the subject of talks presented to engineers of the AiResearch Manufacturing Company in Phoenix, Arizona, on June 10. Mr. Hecht discussed "High Temperature Bearing Development" and Mr. Hanau spoke on "High Temperature Ball Bearing Applications."

Charles F. Becher, spindle development and service engineer in New Departure's Product Engineering Department, addressed the American Society of Machine Tool and Die Manufacturers in Memphis, Tennessee, on June 10. He outlined "The Application and Maintenance of Ball Bearings in Machine Tools, Instruments, and Industrial Equipment."

On June 15, during the Golden Anniversary Summer Meeting of the S.A.E., **L. J. Kehoe, Jr.**, senior project engineer in the GM Engineering Staff's Structure and Suspension Group, presented the paper "A Critical Look at Chassis Lubricants."

A Typical Problem in Engineering

Determine the Basic Design Specifications for a Ball Bearing Screw Assembly

By GEORGE A. WIDMOYER
Saginaw Steering
Gear Division

Assisted by Merle L. DeMoss
General Motors Institute

The ball bearing screw assembly, a revision of the basic power screw principle, finds use in many highly diversified applications. Each assembly must be designed individually to meet specific load, space, and life requirements. To lower manufacturing costs by utilizing existing tooling facilities, the engineer tries to adapt a standard screw design to meet the specifications. This requires that the engineer first must calculate the critical speed, column strength, and bore diameter afforded by the existing design. If the calculations indicate that the existing screw design can be adapted to meet the specific application, the overall basic design specifications for the ball nut are then determined.

Solve the problems of column strength, critical speed, bore diameter, and torque input

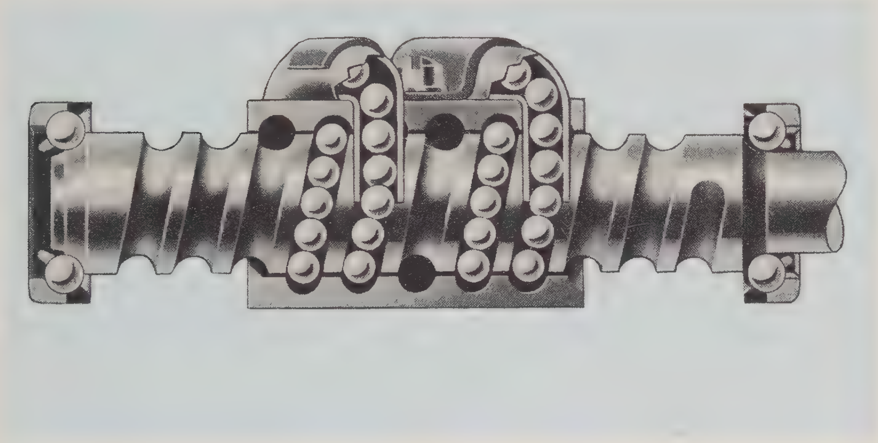


Fig. 1—The ball bearing screw assembly utilizes a screw and a mating nut, each of which has a specially formed, concave, helical ball race. The length of the race within the nut is filled with steel balls of the same diameter. Tubular ball guides interrupt the path of the balls and deflect them from the helical race, then diagonally across the outside of the nut, and back again into the race.

There are many methods used for the transmission of mechanical power or motion. One fundamental method is the power screw which utilizes a screw and nut arrangement to raise a weight or to overcome a resistance. A simple form of power screw, for example, is the familiar screw jack in which either the screw or nut may be stationary while the other is movable.

A revision of the fundamental theory of power screw operation is represented by the ball bearing screw assembly in which the rolling ball principle is applied to a screw and nut to produce a highly effective means for efficiently translating rotary motion to linear motion or linear motion to rotary motion.

The conventional design of the ball bearing screw assembly utilizes a screw and mating nut, each of which has a specially formed, concave, helical ball race (Fig. 1). Within the nut, the length of the helical race is filled with steel balls which act as the medium of engagement between the nut and screw. Each ball is of the same diameter and carries an equal share of the applied load. Fitted to the nut are tubular ball guides which interrupt the path of the balls and deflect them from the helical race. The balls, after being deflected, are guided diagonally across the outside of the mating nut and back again into the helical race.

In many applications a single circuit design—in which one tubular ball guide is used—is adequate. In critical applications where constant or very frequent operation is required the multiple circuit—utilizing two or more tubular ball guides—is used.

The heart of a multiple circuit ball bearing screw assembly is a series of

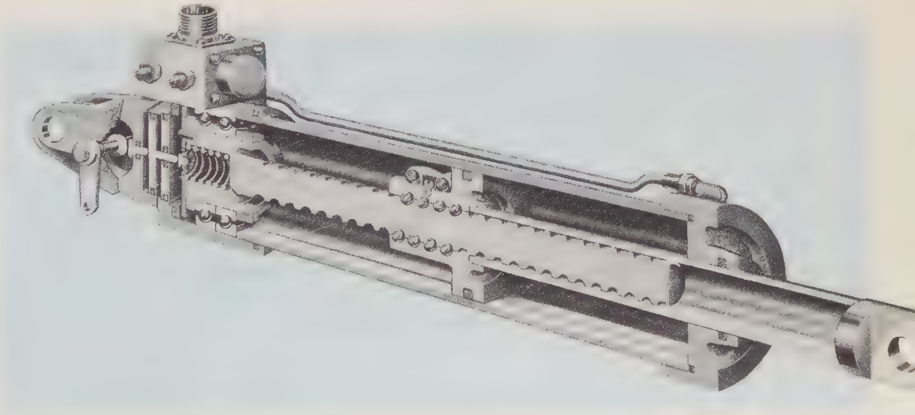


Fig. 2—The ball bearing screw assembly can be combined with either electrical, mechanical, hydraulic, or pneumatic mechanisms. In the example shown, the ball bearing screw assembly, combined with a hydraulic locking cylinder, provides a mechanical locking device which locks the cylinder positively in any desired position. Through direct connection to an indicator (not shown) the assembly provides a constant and accurate indication of cylinder position.

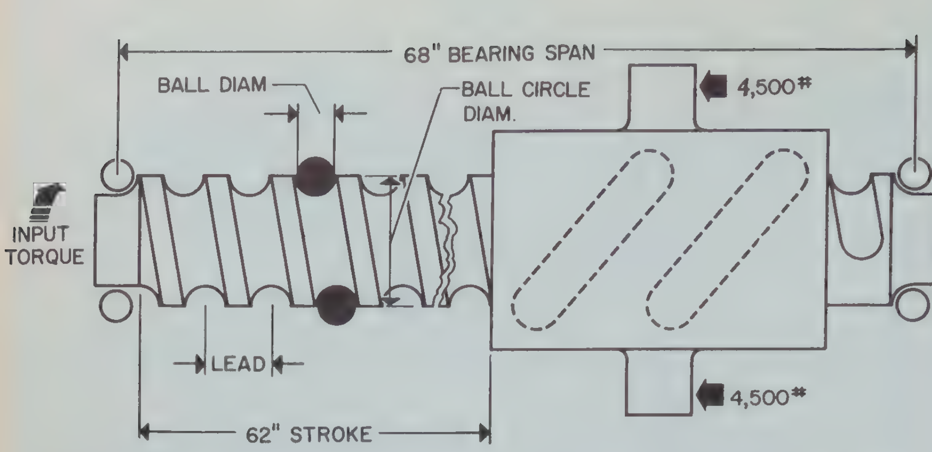


Fig. 3—A 9,000-lb total load is applied to the trunions of the ball nut. The total load is applied equally to each side of the nut to prevent a "cocking" action and to assure that the resultant force is along the longitudinal axis of the assembly. The screw is mounted in radial-type thrust bearings having a span of 68 in. The balls to be used have a diameter of 0.1875 in. The screw has a lead of 0.36363 in., a 1.625-in. ball circle diameter, and an effective column length of 65 in. The screw rotates and actuates the trunion-type ball nut, which does not rotate, on a 62-in. stroke. The dotted lines indicate the position of the tubular ball guides. The number of guides required will be known after the number of ball circuits is determined.

helical races. Each race is an integral part of a closed circuit through which the rolling balls recirculate continually as either the screw or nut rotate relative to each other. Each circuit, the length of which is equal to a specific number of turns, carries a full share of the load and every ball within the circuit does an equal share of the work. If any closed circuit should fail, the remaining circuits continue to function without interruption, with the load being shared among them.

The early application of the ball bearing screw assembly was in the steering mechanisms of passenger cars, trucks, and buses. Since then, many highly diversified applications have been found for use of the assembly outside the automotive industry. As an example, the ball bearing screw assembly is used for actuating landing gears and control surfaces for aircraft and for controlling feed screws and elevating screws on heavy duty grinders. The assembly can be combined with either electrical, mechanical, hydraulic, or pneumatic mechanisms. Fig. 2 shows an example of a ball bearing screw assembly combined with a hydraulic locking cylinder.

Problem

A ball bearing screw assembly is to be used as an actuator for extending and retracting an aircraft's wing flap. The assembly is to have a design load of 9,000-lb tension and compression applied parallel to the centerline of the steel screw. The screw will be the driven member of the assembly with the 9,000-lb total acting through trunnions on the ball nut (Fig. 3). The screw is to rotate at 960 rpm to actuate the ball nut on a 62-in. stroke. The ball nut will not rotate and will be free to move back and forth along the screw's axis. The screw, having an effective span of 68 in., is to be mounted in radial-type, non-rigid thrust bearings. The effective column length of the screw is 65 in. The required life of the ball bearing screw assembly is to be a minimum of 12,000 cycles. One cycle is equal to two strokes of the ball nut.

Assume that an existing screw design has a lead of 0.36363 in., uses 0.1875 in. diameter balls, and has a 1.625-in. ball circle diameter (pitch diameter). For economy of design and production, it is desired to design the assembly to meet the specified load, space, and life requirements by utilizing the existing screw design. To assure that this can be accomplished, it will be necessary first to determine the following:

(a) Determine the column strength of the solid screw to check if it will withstand the design load of 9,000 lb

(b) If the solid screw is overdesigned for column strength, determine the diameter of the bore that can be used for the screw to reduce the overall weight.

(c) Determine the first critical speed of screw rotation to check if the required rotation of 960 rpm is within a safe limit. (This first critical speed is the lowest natural frequency of lateral vibration.)

When calculating (a), (b), and (c), assume the effective outside diameter of the screw to be the root diameter (ball circle diameter minus one ball diameter). Also, consider the screw as a column having both ends pivoted or hinged and free to turn about its axis, but not free to move laterally.

If the values obtained for (a), (b), and (c) indicate that the existing screw design can be used, the following factors then must be calculated to establish the basic design specifications:

(d) Determine the number of ball circuits and the number of turns of balls per circuit to be used in the ball nut.

A ball nut may contain one, two, or three circuits, each of either one and one-half, two and one-half, or three and one-half turns of balls per circuit. When a multiple circuit ball nut is used, each circuit usually contains the same number of turns.

(e) Determine the number of cycles the assembly can be expected to function without ball failure.

The life of a ball is dependent upon the load and the number of impacts the ball makes upon the race as it rolls between the ball thread groove on the nut and screw. The impacts, in turn, equal the number of times a ball passes a given spot in the raceway of the nut. The recommended maximum load for a 0.1875-in. diameter ball is 116 lb per ball for 1,000,000 impacts. The number of impacts per revolution for a 1.625-in. ball circle diameter with 0.1875-in. diameter balls is 11.75.

Established data state that the life of a ball is inversely proportional to the cube of the load on the ball. For example, the allowable load for the 0.1875-in. diameter ball for 1,000,000 impacts is 116 lb per ball. If 8,000,000 impacts would be required, the allowable load would be reduced to 58 lb per ball. ($8 \times 10^6 / 1 \times 10^6 = 8$; $\sqrt[3]{8} = 2$; $116/2 = 58$ lb per ball.)

(f) Assuming the efficiency to be 92 per cent, determine the input torque required to drive the screw under the applied design load of 9,000 lb.

The solution to the problem will appear in the November-December 1955 issue of the General Motors Engineering Journal.

Solution to the Previous Problem

Determine the Interference Fit and Resulting Stresses in the Design of a Cold Extrusion Die

By CHARLES R. BRADLEE
Process Development
Section

Assisted by Robert W. Bund
General Motors Institute

The cold extrusion of a piston pin requires the design of a die consisting of two thick-wall cylinders pressed together to produce an interference fit. The solution to this problem is found using Birne's equation for an open-end cylinder and Timoshenko and Lessell's equation for the pressure between press assembled cylinders. The amount of press required using a 2° taper is 0.128 in. The Brinell hardness number required is 389 Bhn for the outer ring to insure a safety factor of 2 when the working pressure at the bore is 150,000 psi. This is the solution to the problem presented in the July-August 1955 issue of the GENERAL MOTORS ENGINEERING JOURNAL.

Optimum fit of double-wall cylinder permits reasonable stresses

THE need for limiting stresses in metal working tools is not always obvious to the tool designer. In many instances where high stresses are clearly recognizable the antidote is not so obvious. The problem presented in the last issue represents neither extreme but does employ a solution rarely encountered by the engineer or designer.

The total problem of designing the extrusion die (Fig. 1) was necessarily abbreviated to some extent and reduced to the following:

(a) Calculate the distance the two die sections must be pressed together to obtain an initial bore stress of –100,000 psi

(b) Assuming the tensile strength of a steel can be approximated by the formula *Tensile strength = 500 × Brinell hardness number*, what hardness must be specified for the outer ring to insure a safety factor of 2 when the working pressure at the bore is 150,000 psi?

It was suggested that Birne's equation[1] for stresses in a heavy-wall open-end cylinder, and Timoshenko and Lessell's equation[2] for the pressure between two cylinders be used in the solution. In order to proceed in an orderly fashion these equations should be set down in a useful form.

Birne's equation for tangential stress in a cylinder wall under pressure is:

$$S_t = (1 - m)\,\frac{P_i d_i^2 - P_o d_o^2}{d_o^2 - d_i^2} + (1 + m)\,\frac{d_i^2\, d_o^2}{4r^2}\,\frac{(P_i - P_o)}{(d_o^2 - d_i^2)} \quad (1)$$

where

S_t = effective tangential stress
P_i = inside pressure
P_o = outside pressure
d_i = inside diameter of cylinder under consideration
d_o = outside diameter of cylinder under consideration
m = Poisson's ratio (0.30)
r = radius of point for which stress is to be calculated.

In considering the inner cylinder of the assembly shown in Fig. 1, $P_i = 0$; $P_o = P_a$ (pressure due to assembly); $d_i = D_i$ (inside die diameter); and $d_o = D_c$ (common diameter). Equation (1) then can be rewritten for the case of the inner cylinder only as follows:

$$S_t = (1 - m)\,\frac{-\,P_a\, D_c^2}{D_c^2 - D_i^2} + (1 + m)\,\frac{D_i^2\, D_c^2}{4r^2}\left(\frac{-\,P_a}{P_c^2 - D_i^2}\right). \quad (2)$$

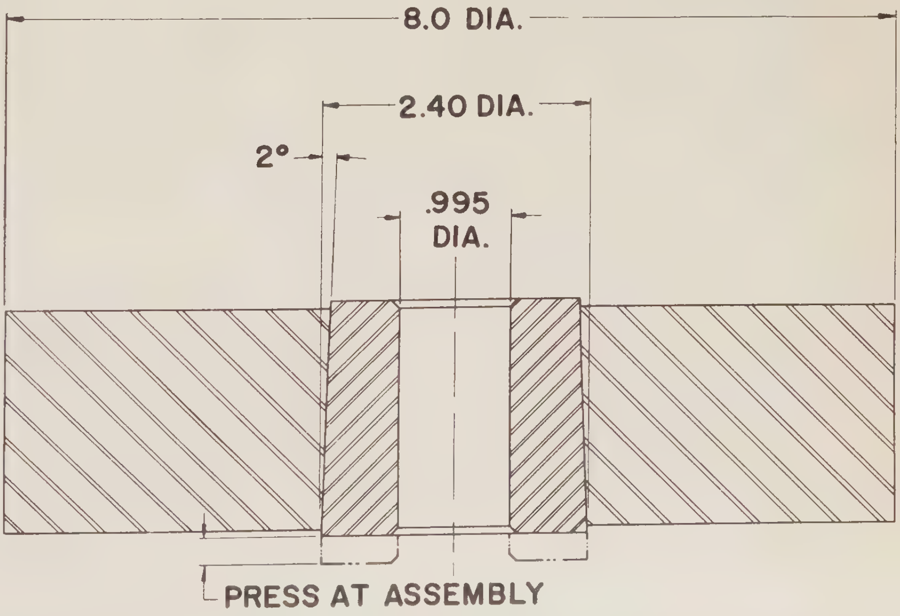


Fig. 1—Shown in this drawing are the principal dimensions and conditions established for the double wall cylinder to be used in the design of an extrusion die. The problem is to calculate the dimension *press at assembly* which will give an initial bore stress of –100,000 psi.

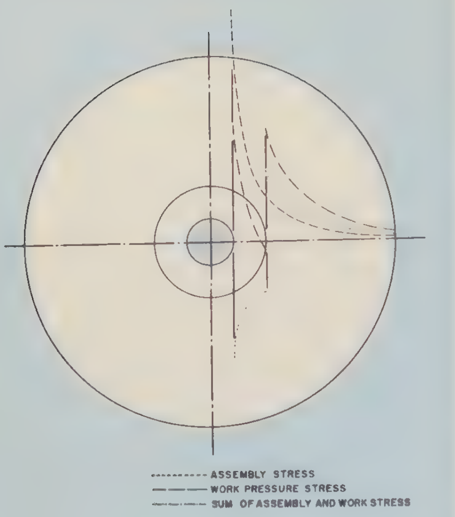


Fig. 2—This diagram shows a graphical representation of stresses in a double-wall cylinder resulting from an interference fit in assembly and when internal pressures are added to the assembly.

Considering only the outer cylinder of the assembly, $P_i = P_a$; $P_o = 0$; $d_i = D_c$; and $d_o = D_o$ (outside die diameter). Equation (1) then can be rewritten for the case of the outer cylinder as follows:

$$S_t = (1 - m)\frac{P_a D_c^2}{D_o^2 - D_c^2} + (1 + m)\frac{D_c^2 D_o^2}{4r^2}\left(\frac{P_a}{D_o^2 - D_c^2}\right). \quad (3)$$

Timoshenko and Lessell's equation for the pressure between two interferring cylinders is:

$$P_a = \frac{EI}{2}\frac{(D_c^2 - D_i^2)(D_o^2 - D_c^2)}{D_c^3 (D_o^2 - D_i^2)} \quad (4)$$

where

E = Young's Modulus

I = total interference between cylinders.

The first part of the problem requires that the distance of press between the tapered sections at assembly be calculated so that the initial bore stress is –100,000 psi. Following is the solution:

Substitute –100,000 for S_t (inner) and $D_i/2$ for r in equation (2) and solve for P_a.

Substitute and factor

$$-100,000 = -\frac{2 P_a D_c^2}{D_c^2 - D_i^2}$$

$$P_a = 41,300 \text{ psi.}$$

Having a value for P_a, values can be substituted in equation (4) and a value for I determined:

Substitute and factor

$$41,300 = \frac{I}{D_c} \times \frac{E}{2} \times \frac{(D_c^2 - D_i^2)(D_o^2 - D_c^2)}{D_c^2 (D_o^2 - D_i^2)}$$

I/D_c = unit interference

= 0.00373 in. per in. of diameter

I = 0.00895 in.

Knowing the total interference and the taper angle of the die sections, the amount of press L_P required can be calculated as

$$L_P = \frac{I}{2 \times \tan \alpha}$$

where

L_P = distance required for press

I = total interference

α = taper angle.

Therefore,

$$L_P = \frac{0.00895}{2 \times \tan 2°}$$

$$L_P = 0.128 \text{ in.}$$

In order to determine the strength required of the outer cylinder, it is first necessary to calculate the value of the maximum stress under working pressure. This will be found at the inner surface of the outer cylinder and will be a total of that resulting from the interference at assembly plus that resulting from the working pressure. Following is the solution:

Substitute the value of P_a, determined above, into equation (3) letting $r = D_c/2$ (the bore radius) and solve for the stress due to assembly.

Substitute and factor

$$S_t = \frac{P_a}{D_o^2 - D_c^2}\left[(1-m)\,D_c^2 + (1+m)\,D_o^2\right]$$

$$S_t = \frac{41,300}{D_o^2 - D_c^2}\left[(1-m)\,D_c^2 + (1+m)\,D_o^2\right]$$

S_t = 61,200 psi due to assembly.

To find the stress increase at the bore of the outer cylinder due to the working pressure, the die is considered as a single cylinder. Values then can be substituted into equation (1) where P_i = 150,000 psi; $P_o = 0$; $d_i = D_i$; $d_o = D_o$; and $r = D_c/2$.

Making these substitutions and factoring, the stress increase due to the working pressure can be obtained as

$$S_t = \frac{P_i D_i^2}{D_o^2 - D_i^2}\left[(1-m) + (1+m)\frac{D_o^2}{D_c^2}\right]$$

$$S_t = 36,000 \text{ psi.}$$

Total stress at bore of outer cylinder during working cycle is

$$S_t \text{ (total)} = 61,200 + 36,000$$

$$S_t = 97,200 \text{ psi.}$$

In order to insure a safety factor of 2 in the strength of the outer cylinder the tensile strength S_t' must be:

$$S_t' = 2 \times 97,200$$

$$= 194,400 \text{ psi.}$$

The Brinell hardness number required to obtain this strength is

$$\text{Bhn} = \frac{S_t'}{500}$$

$$\text{Bhn} = \frac{194,400}{500}$$

$$\text{Bhn} = 389 \text{ Bhn.}$$

Further calculations will show the bore stress in the die under working pressure to be 100,000 psi. Were it not for the initial compressive stress of –100,000 the bore stress would reach 200,000 psi during the work stroke. Fig. 2 shows the various stresses caused by assembly, work pressure, and the sum of the two.

Bibliography

1. Vallance, Alex and Doughtie, V. L., *Design of Machine Members* (New York: McGraw-Hill Book Company, Inc., 2nd Edition, 1943).
2. Eshbach, Ovid W., *Handbook of Engineering Fundamentals* (New York: John Wiley and Sons, Inc., 1936).

Other related literature in this field includes the following:

Seely, Fred B. and Smith, James O., *Advanced Mechanics of Materials* (New York: John Wiley and Sons, Inc., 2nd Edition, 1952).

Timoshenko, Stephen, *Strength of Materials* (New York: VanNostrand, Volume 2, 2nd Edition, 1940-'41).

Engineering Exhibits and Equipment Featured in Chicago Powerama

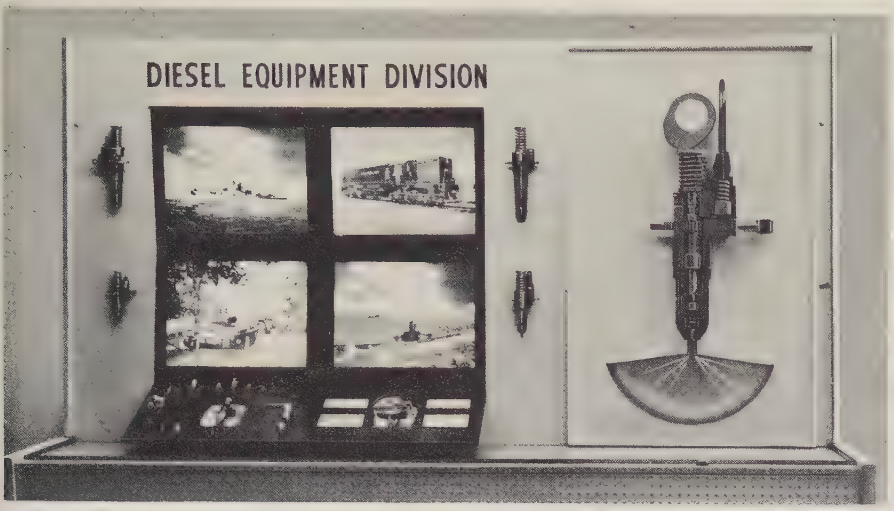


Fig. 1—A typical cutaway exhibit shown at the GM Powerama is this display showing the principal parts and operation of a Diesel unit fuel injector. Manufactured by the Diesel Equipment Division of GM, this injector is a type used on relatively small Diesel engines which furnish power for buses, trucks, and tractors. The display also includes four actual-size injectors which are used in power plants for marine equipment, off-the-road vehicles, and locomotives.

Engineering educators and students are invited to view an array of technical, informative displays in a giant outdoor exposition being held in Chicago, August 31 through September 25. Called Powerama, the exposition is presented by General Motors to portray the dramatic contributions of Diesel and aircraft power to America's industrial economy. It is open to the public at no admission charge on the site of the 1933-34 Century of Progress Exposition, adjacent to Soldier Field on South Lake Shore Drive.

On display is a newly designed train built by the Electro-Motive and GMC Truck & Coach Divisions of General Motors at the request of railroad officials to explore the possibilities of a lightweight, low-cost, Diesel powered passenger train as a means of reducing operating costs.

One of the technical exhibits is a working demonstration of an aluminum foundry which actually produces a casting for Diesel locomotive engines. Also in the foundry exhibit is a demonstration of making patterns by the lost wax process, plus other displays demonstrating a permanent mold operation and investment mold operation, sand cast molding and die cast molding.

A road building exhibit illustrates the steps in building a modern highway and another exhibit will allow visitors to get the "feel" of operating heavy duty off-the-road equipment. The latter exhibit compares the performance of an Allison Torqmatic Drive with a mechanical transmission.

Other equipment displayed includes a shrimp boat, a pleasure launch, and a U.S. Navy submarine—all of which can be boarded by visitors and which demonstrate various applications of Diesel engine power.

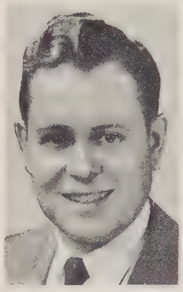
At another exhibit demonstrating a typical precision manufacturing process for Diesel engine components, an operator drills fuel spray holes in the spray tip of a unit injector which, in production models, range in diameter from 0.017 in. down to 0.005 in.

Powerama also includes numerous cutaways demonstrating construction details and engineering principles of such products as gas turbine engines, transmissions, aircraft propeller controls, and various engine components. A typical exhibit shows the parts of a Diesel engine unit injector (Fig. 1).

The GM Divisions contributing to Powerama include Allison, Cleveland Diesel Engine, Detroit Diesel Engine, Diesel Equipment, Electro-Motive, Euclid, Fabricast, (known in General Motors as the engine group), as well as Frigidaire and GMC Truck & Coach Divisions.

Contributors to Sept.-Oct. 1955

GENERAL MOTORS
Engineering
JOURNAL



EDWARD L. BEERS, contributor of "Applying Material Handling Principles to Containers in the Manufacture of Appliance Parts," serves Delco Appliance Division, Rochester, New York, as a material handling engineer in its Production Control Department.

In June 1943 he joined this Division's Product Engineering Department for summer employment as a blueprint clerk. One year later, he rejoined this same Department on a full-time basis as a draftsman. In the fall of 1944 Mr. Beers enrolled in the engineering program at General Motors Institute, Flint, Michigan, graduating in August 1950. During this six-year period he was on military leave of absence from May 1945 until December 1946, serving in the U.S. Army.

After graduation from G.M.I., Mr. Beers was assigned to the Plant Layout Department at Delco Appliance where he compiled and wrote his Fifth-Year Project study report entitled "The Improvement of Material Handling Methods and Equipment." This report led to Mr. Beers earning the Bachelor of Industrial Engineering degree from General Motors Institute in 1951.

In December 1952 Mr. Beers transferred to the Production Control Department where he is concentrating on material handling problems connected with warehousing, raw stores, traffic, receiving, and packaging.

Mr. Beers is a charter member and a past vice president and director of the Rochester Chapter of the American Material Handling Society.

CHARLES R. BRADLEE, who prepared the problem "Determine the Interference Fit and Resulting Stresses in the Design of a Cold Extrusion Die" and the solution appearing in this issue, is a project engineer in the Engineering Department of the Process Development Section.

The Section, which is located at the General Motors Technical Center just north of Detroit, is a centralized activity of the General Motors Manufacturing Staff, established to supplement divisional engineering facilities in the development of improved manufacturing techniques and processes. Since its inception in 1946, the Section has developed some new and unusual equipment for manufacturing, assembly, and inspection.

Mr. Bradlee began his General Motors career with this organization in August 1952 as a junior project engineer. A year later he was promoted to his present position of project engineer. He is currently engaged in developmental work on various aspects of cold forming techniques. This process enables shaping operations to be accomplished with metals at temperatures below their recrystallization point. Among the benefits to be derived from this process are product quality improvement, material savings, and increased physical strength.

Illinois Institute of Technology granted Mr. Bradlee the B.S. degree in 1949. Following graduation, he was employed for three years by the General Cable Corporation, New York, New York.

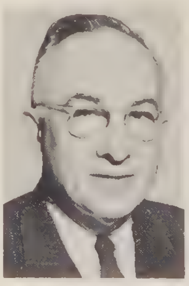
Mr. Bradlee's professional affiliations include the Society of Automotive Engineers, the American Society of Mechanical Engineers, and the honorary societies Pi Tau Sigma and Tau Beta Pi.

PHILIP N. BRIGHT, contributor of "Structural Design Problems in Gas Turbine Engines," is group project engineer, structural design, in the Power Turbine Engineering Department of Allison Division, Indianapolis.

Mr. Bright received the B.S. degree in mechanical engineering (aeronautical) from the Carnegie Institute of Technology in 1937. He spent the next six years performing flight test engineering and stress analyses work for the Glenn L. Martin Company, Baltimore, Maryland. His General Motors career began in February 1943 as project structural engineer with Fisher Body Division's Aircraft Development Section. He left GM the following year to become chief structures engineer for, first, Aeronca Aircraft Corporation, Middletown, Ohio, and, later, Southern Aircraft Corporation, Dallas, Texas.

In 1946 he returned to General Motors as structures engineer at Allison Division. Shortly after, he was promoted to his present position which currently involves structural design of the T56 and other advanced turbo-prop engines. Previously, Mr. Bright conducted developmental structural design on the T38 and T40 turbo-prop engines.

Mr. Bright occupies the dual positions of chairman of the Indianapolis Sections of both the Institute of the Aeronautical Sciences and the Society for Experimental Stress Analysis during the 1955-56 season.



C. A. BROWN, co-contributor of "How to Organize and Write Effective Technical Reports," serves as administrative assistant in charge of instruction and faculty development in the English and Psychology Department at General Motors Institute, Flint, Michigan. Mr. Brown has held this position since February 1955. He previously served as chairman of that Department, a post which he had occupied from the time he joined the faculty of the Institute in September 1929.

Purdue University awarded Mr. Brown the B.S. degree in 1924, following which he taught at that University for five years. In 1929 he earned the A.M. degree from the University of Chicago.

He is a member of Alpha Tau Iota, the American Society for Engineering Education, the National Council of Teachers of English, the College English Association, and the American Business Writers Association. In addition, Mr. Brown is professionally active in the A.S.E.E., serving as chairman of the Michigan Section and chairman of the English Division on the General Council for the past six years.

His public presentations have included papers on technical writing before several engineering societies. Among material published by Mr. Brown have been many articles in the *Journal of Engineering Education.* A book entitled *Writing Useful Reports*, which he is co-authoring with R. E. Tuttle, also of the Institute faculty, will be published this fall by Appleton-Century-Crofts.

Mr. Brown is a veteran of the first World War, having served in the Medical Corps for two years, and is listed in *Who's Who in Engineering* and *Who's Who in American Education.*

DONALD P. BUSWELL, co-contributor of "A Discussion of Economic Factors Affecting the Steel Selection and Heat Treatment for Automotive Gears," is plant metallurgist in the Metallurgical Department of Buick Motor Division's Manufacturing Division, located in Flint, Michigan.

Mr. Buswell joined this Department in 1942 as a metallurgist, directly after earning the B.S. degree from the University of Wisconsin. In 1944 he was made experimental metallurgist. From April 1946 to April 1947 he was on military leave, serving in the U. S. Army Counter Intelligence Corps. In 1950 he was assigned to Buick Motor's Rear Axle Plant for one year. Then he was made assistant plant metallurgist and, in 1953, was promoted to his present position.

As plant metallurgist, Mr. Buswell is responsible for the plant control of metallurgical processes and for the supervision of all metallurgical inspection. Previously, Mr. Buswell did work on various National Research Council projects.

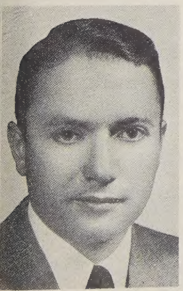
Mr. Buswell is a member of Tau Beta Pi and his technical memberships include the American Society for Metals and the General Motors Heat Treat Subcommittee.

EDWIN S. DYBVIG, contributor of this issue's "Notes About Inventions and Inventors," is a patent attorney in the General Motors Patent Section, assigned to its Dayton Office, Dayton, Ohio. This office is one of three principal locations of patent activities in General Motors; other offices are at Washington, D. C., and at the GM Central Office in Detroit, Michigan. The Dayton Office handles patent, copyright, and trademark matters, as well as licensing agreements affecting a number of GM Divisions, including those located in the Dayton area.

In 1929 Mr. Dybvig earned the electrical engineering degree from the University of Minnesota, and in 1933 obtained the LL.B. degree from Washington College of Law, Washington, D. C. While working toward the latter, he was engaged as an examiner in the United States Patent Office. He remained there until 1936 when he went to General Motors as a patent attorney in Frigidaire Division's Patent Section

Mr. Dybvig left General Motors in 1947 to establish a family patent law firm in Dayton. After five years of private practice, he returned to GM where he assumed his present post. Mr. Dybvig's major responsibility is to handle patent problems in the field of air conditioning and refrigeration. Four patents have been granted in his name as a result of his survey work in this area.

He is a member of the bar of the State of Ohio and a member of the District of Columbia Bar. In addition he serves on the American Patent Law Association's Committee on Relations with the Patent Office and is currently president of the Dayton Patent Law Association. He is also a member of the Engineers Club of Dayton.



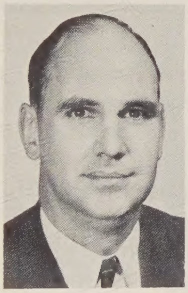
CARL L. GOODZEIT, co-contributor of "Why Bearings Seize," started with the Research Laboratories Division in June 1951 as a college graduate-in-training. Two years later—November 1953—he was promoted to research engineer in the Division's Mechanical Development Department, his present position.

Mr. Goodzeit holds both the B.S. degree in mechanical engineering (Rutgers University, 1950) and the M.S. degree in engineering (Brown University, 1955).

Mr. Goodzeit's past projects have included work in the fields of lubrication, friction, wear, and automatic transmissions. Presently, he is concerned with research on bearing materials—the general area of his paper appearing in this issue.

He has presented a paper on wear before a meeting of the American Society of Lubrication Engineers and on separate occasions has presented papers on bearing materials and on the friction of sliding metals before the American Society of Mechanical Engineers.

In addition, Mr. Goodzeit's technical affiliations include membership on the A.S.M.E. Lubrication Committee's Subgroup on Boundary Lubrication and Bearing Materials.



VERNON E. HENSE, co-contributor of "A Discussion of Economic Factors Affecting the Steel Selection and Heat Treatment for Automotive Gears," is chief metallurgical engineer in the Metallurgical Department of Buick Motor Division's Manufacturing Division, Flint, Michigan. His major concern is the supervision of those metallurgical functions of the Manufacturing Division affecting the Buick automobile and the Wright J-65 Sapphire jet engine.

Mr. Hense is a 1934 automotive engineering graduate of General Motors Institute. In 1930 he joined Buick Motor Division, Flint, as a G.M.I. co-op student and, four years later, transferred to the Buick Motor Transmission Plant as a metallurgist. His promotions since then include: assistant plant metallurgist (1936), plant metallurgist (February 1941), assistant chief metallurgical engineer (September 1941), and chief metallurgical engineer—his present position (1951).

Contributors' backgrounds vary greatly in detail but each has achieved a technical responsibility in the field in which he writes.

Mr. Hense's wartime projects included metallurgical work on the M-4 tank transmission, the M-18 Hell Cat tank, the 75 mm steel cartridge case, several National Research Council projects, and Flint operations on the Pratt and Whitney aircraft motor.

Mr. Hense is a co-author of a paper on "The Economics of Automotive Gear Steels and Their Heat Treatment," presented at the 1950 Society of Automotive Engineers Summer Meeting. This paper discusses earlier practices in the same area as his current paper.

Mr. Hense is active in several technical societies and committees. He is chairman of the Panel C—Automotive, Society of Automotive Engineers; is vice chairman of the Saginaw Valley Chapter of the American Society for Metals; is past chairman of the General Motors Metallurgical Committee; and a member of the American Foundrymens Society.

FRANK V. KLCO, contributor of "The Theoretical and Practical Aspects of Automotive Brake Design and Testing," serves Oldsmobile Division in Lansing, Michigan, as a senior layout man in the Product Engineering Department. He was promoted to this position in November 1952.

In July 1941 Mr. Klco joined AC Spark Plug Division in Flint, Michigan, as an apprentice. He enrolled at General Motors Institute in the fall of that same year. In early 1942 he transferred to Oldsmobile Division.

The second World War interrupted Mr. Klco's GM career at this point while he served in the Air Force from August 1942 until October 1945. Mr. Klco's military occupational specialty was navigator on B-17 bombers and he separated with the rank of first lieutenant.

In 1945 he returned to his studies and was graduated from G.M.I. in August 1948. Mr. Klco's paper in this issue is an adaptation of his thesis on brake design which led to the granting of his B.M.E. degree from G.M.I. one year later. After his graduation from G.M.I. in 1948, he was made a detailer in the Product Engineering Department of Oldsmobile. Three years later—March 1951—he was promoted to layout man and was made a senior layout man—his present position—in November 1952. Currently, Mr. Klco is concerned with chassis design work.

Mr. Klco is a member of the Society of Automotive Engineers.

RALPH A. RICHARDSON, co-contributor of "How to Organize and Write Effective Technical Reports," is head of the Administrative Engineering Department of the Research Laboratories Division, located at the General Motors Technical Center. His Department provides engineering services and correlates work of the Laboratories with engineering departments of GM manufacturing Divisions and central Staff operations.

Mr. Richardson was graduated from the University of Minnesota in 1927 with the B.S. degree in mechanical engineering. At this time he joined General Motors as a junior engineer at AC Spark Plug Division in Flint, Michigan, where he worked as a tool and die designer. In the fall of 1927 he was transferred to the Research Laboratories Division and assigned to the Technical Data Department. He became assistant department head in 1932 and four years later was named head of the Department. The Department is now the Administrative Engineering Department.

Mr. Richardson has been a frequent contributor to technical literature. A series of engineering booklets which he prepared for his Department now is used in educational institutions, training courses, and engineering public relations activities. He contributed to the automotive section of Kent's Mechanical Engineers' Handbook and was co-author of a book on the automotive industry published by the Encyclopedia Britannica. He is currently engaged in the editorship of the forthcoming Automotive Engineers' Handbook.

Mr. Richardson's technical affiliations include membership in the Society of Automotive Engineers and the Engineering Society of Detroit. From January 1943 to January 1946 he served in the Navy as a specialist electronics officer and attained the rank of lieutenant commander.

ARVID E. ROACH, co-contributor of "Why Bearings Seize," was supervisor of bearing development in the Research Laboratories Division's Mechanical Development Department from July 1953 until his death in July 1955.

Mr. Roach joined the Division in 1940 as a summer student and upon graduation in 1941 returned to the Research Laboratories as a junior engineer in the Technical Data Department. In 1948 he transferred to the Mechanical Development Department where he concentrated on research in bearings and lubrication. He was recently concerned with the fundamentals of friction and wear and the score resistance of bearing metals—the subject of this, his final contribution to the engineering literature.

Mr. Roach earned the B.E.E. degree from University of Delaware, Newark, in 1941 and the M.S. degree in mechanical engineering from Wayne University, Detroit, in 1949.

From 1944 to 1946, Mr. Roach served in the U.S. Navy, separating with the rank of lieutenant, junior grade.

Mr. Roach completed numerous speaking engagements before various technical societies—on lubrication, bearing design and materials, the theory and design of centrifugal pumps, applied hydraulics, and industrial mathematics.

Mr. Roach served as chairman of the American Society of Lubrication Engineers' Technical Committee on Bearings and Bearing Lubrication and of the A.S.L.E. Research Group—Lubrication Activity. At the time of his fatal illness, he was vice president of the Industrial Mathematics Society and was affiliated with several honorary societies, including Tau Beta Pi and Sigma Xi.

The General Motors Engineering Journal is a publication designed primarily for use by college and university educators in the fields of engineering and the sciences. Educators in these categories may, upon request, be placed on the mailing list to receive copies regularly. Classroom quantities also can be supplied regularly or for special purposes, upon request to the Educational Relations Section, General Motors.

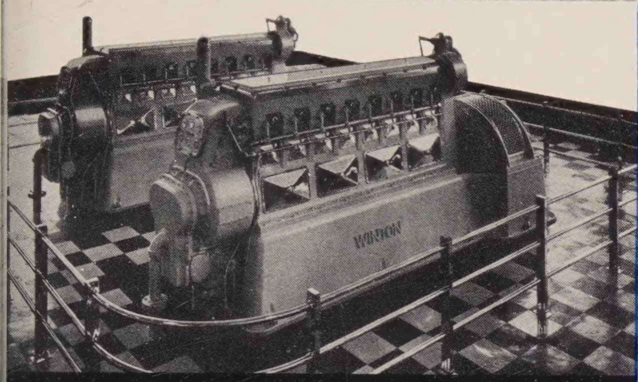
PATON M. ZIMMERMAN, contributor of "Toroidal-Type Current Meter Improves Weld Quality," is senior process engineer in the Process Development Activity of Fisher Body Division, located at the GM Technical Center.

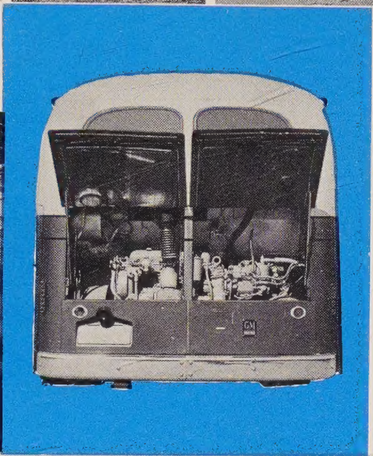
In September 1942, Mr. Zimmerman joined GM's Eastern Aircraft Division in Linden, New Jersey, as a General Motors Institute co-op student. (Eastern Aircraft Division was set up on a temporary basis during World War II to consolidate several of General Motors East Coast plants with the objective of manufacturing complete fighter planes and torpedo bombers for the Navy.)

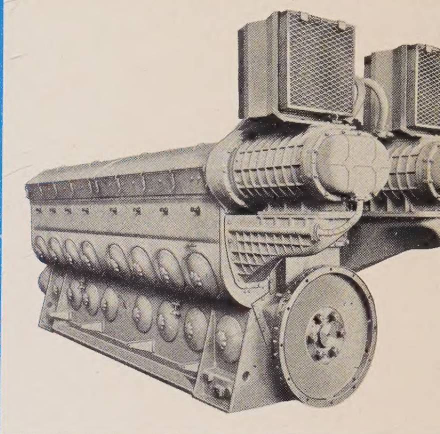
Four years later, in August 1946, Mr. Zimmerman transferred to the Buick-Oldsmobile-Pontiac Assembly Division as a junior process engineer. In this same capacity he transferred to Fisher Body's Process Development Activity in September 1947. Subsequent promotions at this Division have included process engineer and senior process engineer.

Mr. Zimmerman's current work is concerned with technical publicity for the Process Development Activity and the direction of a program for the use of industrial, closed circuit television. Previously, he prepared the *Fisher Welding Guide* for plant welder maintenance personnel and, in cooperation with General Motors Institute, set up an associated training program.

Mr. Zimmerman received the B.M.E. degree from General Motors Institute in 1947 and is a member of G.M.I.'s honorary engineering society, Alpha Tau Iota. He is a member of the American Welding Society and his committee work includes serving as chairman of the Working Committee for the Fisher Body Advisory Committee on Noise Reduction.



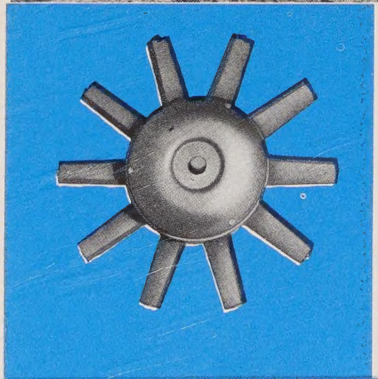


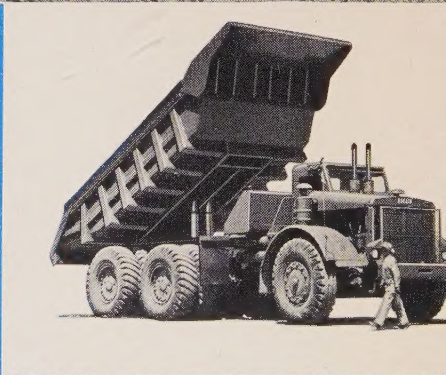


DIESEL POWER

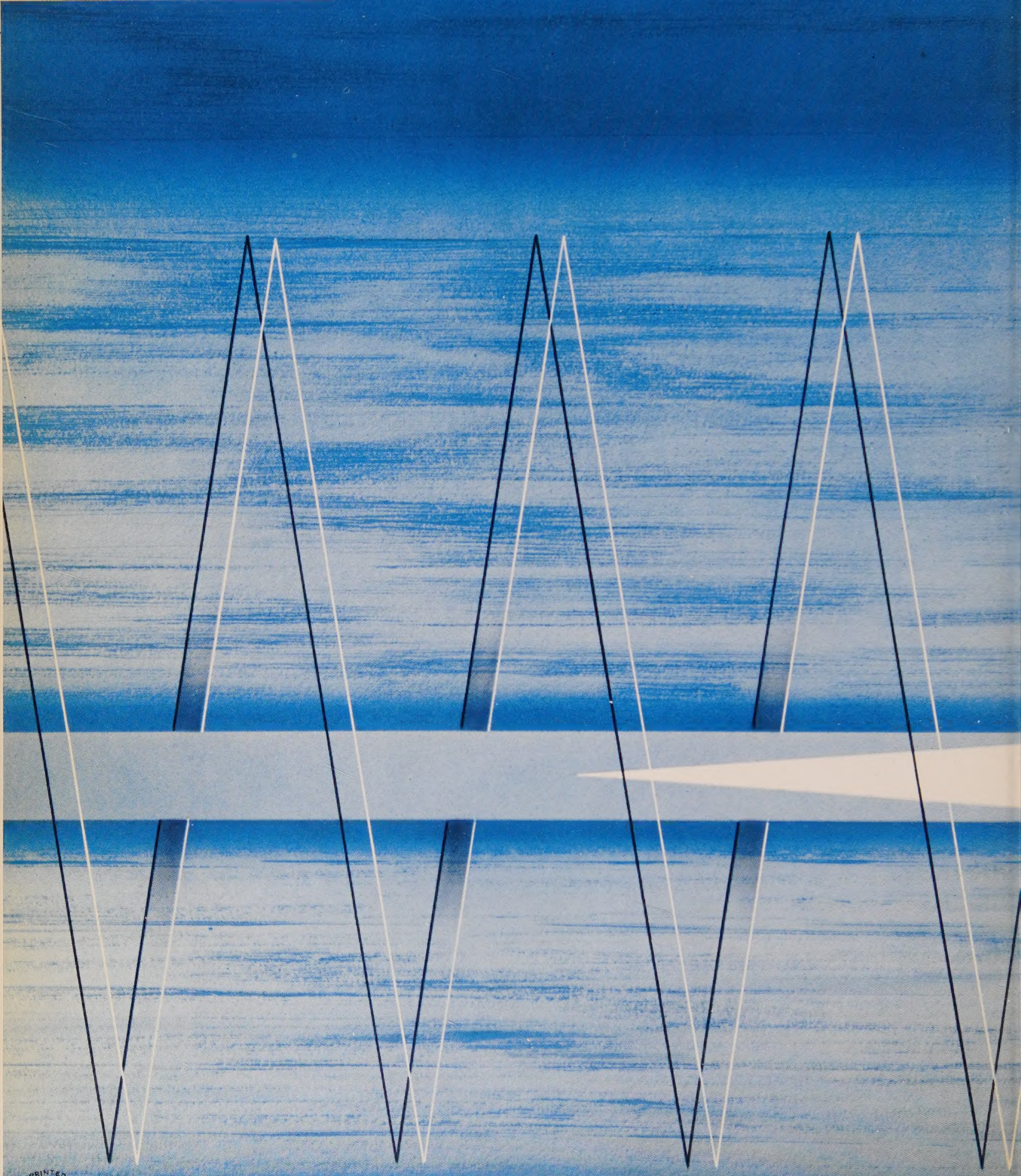
Early Diesel engines were bulky, heavy, and slow speed because they were designed according to steam engine practice and built to steam engine standards. For example, successful Diesels built in the 1920's weighed up to 200 pounds per horsepower and revolved at average speeds of only 150 rpm to 600 rpm, yet they were efficient, dependable, and inexpensive to maintain. If the Diesel was to be successful in the motive power field it had to be made lighter and had to turn faster. The development of a practical unit injector, a suitable helical-type blower, specially designed pistons, bearings, and many other basic engine components led to the design of the first successful high-speed, two-cycle, Diesel engine by GM's research engineers. In cooperation with the Winton Engine Company—which became a part of General Motors in 1930—GM Research Laboratories Division's engineers built a pair of two-cycle Diesels rated at 600 hp each to generate electric power for the GM exhibits at the 1933 Chicago World's Fair. These engines created intense interest among railroad men and other potential users of Diesel power. The result was an entire new industry, new products, and new jobs for thousands of people. From this single engineering accomplishment grew the following organizations in General Motors:

- **Cleveland Diesel Engine Division**—engines for marine and industrial uses
- **Detroit Diesel Engine Division**—engines for trucks, buses, tractors, earthmoving equipment, well-drilling rigs, generator sets, and other industrial applications
- **Diesel Equipment Division**—fuel injectors, hydraulic valve lifters, gas valves, piston rings, and other engine components
- **Electro-Motive Division**—locomotives, heavy-duty mobile generating plants, and oil well drilling





GENERAL MOTORS
Engineering
JOURNAL



PRINTED IN